VENTILATION OF ELECTRICAL MACHINERY

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W. H. F. MURDOCH

THE VENTILATION OF ELECTRICAL MACHINERY

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BY

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WITH THIRTY ILLUSTRATIONS

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PREFACE

This little work is intended as an introduction to the problem of ventilation applied to electrical machinery.

At present the cost of electric generators is high, partly on account of the fact that only a small temperature rise is permissible. There seems to be little reason to doubt that with the advance of technical applications and artificial local cooling the cost of large generators may be reduced to one-half, or one-third of the present cost per kilowatt.

Indirectly the question of efficiency is raised through the use of cooling devices, and this is discussed.

Very little appears to have been written on the subject of ventilation, but where the author is indebted to other writers it is duly acknowledged in the text.

Methods of measuring the "ventilating effect" of armatures are indicated in the chapter on "Ventilating Action of Armatures." They may be recommended as rapid workshop tests, and although exception may be taken to them on various grounds they are preferable to the present state of matters, viz. no tests at all.

The author considers the present is a very opportune time to draw attention to this question. It is not generally realized, for instance, that the so-called "Output Coefficients" are merely functions of ventilation, and that it is not correct in discussing them to ignore ventilation altogether.

Meanwhile there appears to be in certain quarters an extraordinary aversion to artificial cooling devices. This seems to arise through an entire misconception regarding the reason for using them, and also, perhaps, because their effect on the efficiency is not quite obvious.

It is hoped these few chapters will tend to dissipate some of these ideas, and, however imperfect they may be, perhaps they may lead others to interest themselves in the matter.

The author is indebted to Messrs. Mavor and Coulson, The British Thomson-Houston Co., and Messrs. H. Wright and Wood for some information regarding their enclosed motors. He has also to thank the James Keith and Blackman Co. for some particulars regarding their fans.

W. H. F. M.

London, 1911.

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THE VENTILATION OF ELECTRICAL MACHINERY

Efficiency of Generators, etc.

Much attention has been given to designing electrical apparatus, enabling it to perform its work in the most efficient manner possible. Very large generators have an efficiency of 93 % and transformers 98 %. This is, of course, a remarkable achievement, and reflects great credit on the designers of such apparatus.

Nevertheless it seems to the author that a great deal remains to be done to enable the same output to be obtained from a cheaper machine by means of more efficient ventilation than is at present used. At present there seems to be a feeling that if one overruns a machine, its efficiency will be greatly reduced, but this is by no means the case.

For instance, if we call O the output of any machine, L the constant losses, excitation, iron, friction, etc., R the armature resistance, C the armature current, and I the input, then we have :—

$$\epsilon_1 = \frac{O}{I}$$

or

$$\epsilon_1 = \frac{O}{O + L + C^2R}.$$

Now suppose that the current is increased k times, the output will now be kO and we have for the efficiency :—

$$\epsilon_2 = \frac{kO}{kO + L + k^2C^2R}$$

or

$$\epsilon_2 = \frac{O}{O + \frac{L}{k} + kC^2R}.$$

Therefore

$$\frac{\epsilon_1}{\epsilon_2} = \frac{\dfrac{O}{O + L + C^2R}}{\dfrac{O}{O + \frac{L}{k} + kC^2R}}$$

$$\frac{\epsilon_1}{\epsilon_2} = \frac{O + \frac{L}{k} + kC^2R}{O + L + C^2R}.$$

Hence $\epsilon_1(O + L + C^2R) = \epsilon_2\left(O + \frac{L}{k} + kC^2R\right).$

If $$\frac{L}{k} + kC^2R > L + C^2R,$$

then $$\epsilon_1 > \epsilon_2.$$

Again, if we write :—

$$L + C^2R = \frac{L}{k} + kC^2R$$

we have

$$k^2 - k.\frac{L + C^2R}{C^2R} + \frac{L}{C^2R} = 0,$$

$$\therefore \quad k = \frac{L + C^2R}{2C^2R} \pm \frac{1}{2}\sqrt{\left(\frac{L + C^2R}{C^2R}\right)^2 - \frac{4L}{C^2R}}.$$

The terms inside the bracket are equal to

$$\frac{L^2 + 2LC^2R + C^4R^2 - 4LC^2R}{C^4R^2}$$

or $$\frac{L^2 - 2LC^2R + C^4R^2}{C^4R^2};$$

this is a perfect square so that

$$k = \frac{L + C^2R}{2C^2R} \pm \frac{L - C^2R}{2C^2R},$$

taking the positive value we have

$$k_1 = \frac{2L}{2C^2R} \text{ or } \frac{L}{C^2R};$$

and $$k_2 = 1,$$

we see then k can only be greater than unity if $L > C^2R$. In many cases this is so; in one particular example $L = 10C^2R$ for a single-phase alternator.

Of course in dynamos the inherent troubles of regulation will limit the application of these calculations, but in alternating machinery a considerably increased output might be obtained provided the heat is satisfactorily removed. The question, then, as to whether an overload will improve the efficiency, or otherwise, depends on the ratio of the constant losses to the variable ones.

Writing

$$k^2 - k \cdot \frac{L + C^2R}{C^2R} + \frac{L}{C^2R} = 0.$$

Differentiating this expression, and equating to zero, we find the value of k corresponding to the vertex of the parabola—

viz. $$k = \frac{L + C^2R}{2C^2R}.$$

Suppose $L = 2C^2R$, we then have the parabola for k as shown in the figure on p. 4. We see that between the points $k = 1$, $k = 2$, the efficiency is greater than the value at starting, and has its maximum value for

the value of k given by the vertex of the parabola, viz. $1\frac{1}{2}$. As a rule overrunning a machine may reduce its efficiency slightly, and with the ordinary type of generator doubtless causes a rise of temperature which might prove objectionable or destructive unless there was some means at hand of getting rid of it.

If the heat generated by the losses could be efficiently

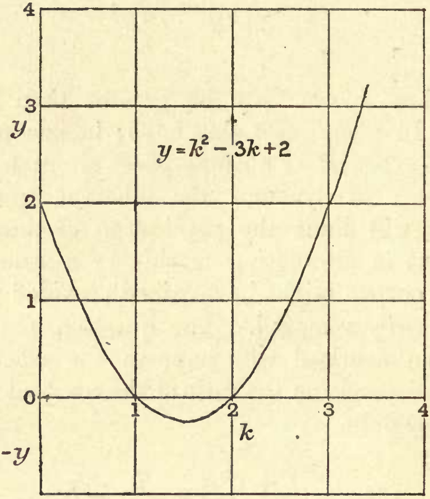

FIG. 1.—Values of k.

got rid of, there would be no limit to the output of say an alternator or dynamo, provided that in the latter case the sparking limit could be kept down.

Meanwhile the thermal limit to the outputs of electrical machinery is one of the greatest drawbacks the electrical engineer has to contend with. Owing to this he has only got a 40° C. rise of temperature to work upon, and at the present time the output of generators limit the output of the turbine in a turbo-generator

set. For this reason the capital cost of the electrical equipment of the station is very heavy, and if some means of running the generating or other plant at higher outputs continuously could be arranged it would undoubtedly lower capital cost and diminish the price charged for electrical energy.

Another disadvantage of high temperatures is that the resistance increases with temperature rise.

$$R_t = R_0(1 + 0{\cdot}004\ t^\circ)$$

where t° is the temperature rise.

Hence for a 40° C. rise the ratio

$$\frac{R_t}{R_0} = 1{\cdot}16,$$

or the C^2R loss is increased by 11·6 %, due to temperature rise.

If the total losses in a machine are written

$$T = L + C^2R_0(1 + 0{\cdot}004\ \theta^\circ)$$
$$\frac{dT}{d\theta} = C^2R_0 \times 0{\cdot}004,$$

or the rate of increase of loss with temperature rise is :—

$$\frac{1}{250}\ C^2R_0,$$

for a copper resistance.

As already mentioned, no doubt hysteresis, etc., increase with temperature rise, but as these changes are small we shall neglect them.

It appears, then, that so far as change of temperature with load is concerned, we need only consider the copper losses. If, then, we can keep down the temperature rise we reduce the copper loss directly, and if we can, by means of using some cooling device, carry off the heat

as fast as it is generated, then the output might be correspondingly increased.

We might write for rate of rise of temperature of any machine with time:—

$$\frac{d\theta}{dt} = L + C^2R_0(1 + a\theta) - S\theta,$$

where S is some effective surface from which the heat escapes. In an enclosed motor this would be the exposed surface of the case of the machine.

If $\frac{d\theta}{dt} = 0$, so that the machine is at some steady temperature, then

$$\theta^\circ = \frac{L + C^2R_0}{S - aC^2R_0}.$$

We see, therefore, that the effect of using material to construct coils which has a high value of a, or increase of resistance per 1° C. per unit resistance, is simply to reduce the effective ventilating surface.

Since, then, the temperature rise has a considerable effect on the efficiency and on the working of a machine, it will be advisable to consider the various factors in more detail.

Assume that a machine is heated by its losses as previously, and that it is cooled by natural ventilation as well as by a fanning or ventilating action depending on the speed. We then have for a general equation:—

$$M\frac{d\theta}{dt} + k_1\theta + k_2n\theta = C^2R_0(1 + a\theta) + Hn + En^2.$$

Here M is the mass of the machine multiplied by its specific heat, k_1, k_2 are constants, n is the speed, H and E the hysteresis and eddy current loss constants respectively.

Integrating this equation we find:—

$$\theta^\circ = \frac{C^2R_0 + Hn + En^2}{k_1 + k_2n - aC^2R_0}\left\{1 - \epsilon - \frac{(k_1 + k_2n - aC^2R_0)t}{M}\right\}.$$

Now k_1 and k_2 are obviously surface factors, and it is interesting to notice the only effect of M is to prevent rapid alterations in temperature of the machine as a whole.

In the above the word temperature has been used, but it must be recollected that the temperature of a machine is a very vague term. We might assume, however, that we are referring to the temperature of the copper winding in the armature, say, or the average temperature of the copper in the armature.

If we assume a steady temperature has been reached, and neglect the terms involving hysteresis and eddies, we might write out the equations for change of temperature with speed.

Writing the equation in the form :—

$$\theta_{\text{max.}} = \frac{C^2R}{S(k_1 + k_2n)},$$

we have :—

$$\frac{k_2}{k_1}n\theta + \theta = \frac{W}{k_1S}.$$

Obviously this curve is a hyperbola, having for asymptotes

$$n = -\frac{k_1}{k_2}$$

and $$\theta = 0,$$

as shown in I of the figure on p. 8.

Again the case where eddies are important we find :—

$$\theta_m = \frac{W}{S(k_1 + k_2n)} + \frac{H}{k_2} + \frac{En}{k_2} \text{ approximately.}$$

And the curve connecting θ and n is a parabolic equation of the type:—

$$(n - A)^2 = B\theta - \text{constant}.$$

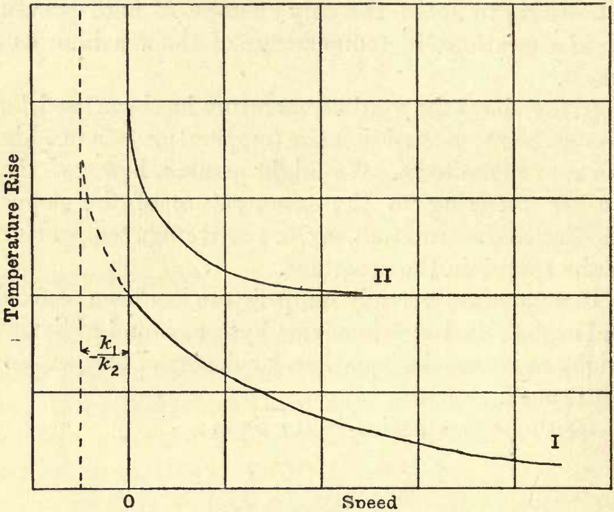

Fig. 2.—I, ventilated machine; II, enclosed machine.

The equation for a totally-enclosed machine is of some interest, and we now proceed to discuss it.

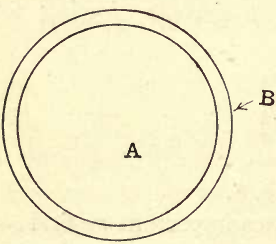

Fig. 3.

In Fourier's treatise on the *Analytical Theory of Heat* it is shown that the heat necessary to keep a body A, at some temperature θ° when surrounded by a narrow air space, is only one-third of that required when it is freely exposed to the atmosphere.

By reasoning in an exactly similar manner we are able to deduce the equation

for variation of temperature with speed of a totally-enclosed machine.

Let θ_1 be the temperature of the air inside the motor case,

θ_2 the inside temperature of the surface of the motor case,

θ_3 the outside temperature of the surface of the motor case,

θ_4 the temperature of the air outside.

Let s, t, ϵ and H, represent the outside surface of the case, the thickness (t) being supposed small, the conductivity of the case and heat per second, and let k_1 and k_2 be constants.

Then, following Fourier's method but modifying it by supposing the air to flow with some velocity v depending on armature speed over the inside of the case, we have :—

$$\theta_1 - \theta_2 = \frac{H(1 + k_1 v)}{Svk_2},$$

$$\theta_2 - \theta_3 = \frac{Ht}{k_3 S},$$

$$\theta_3 - \theta_4 = \frac{H}{\epsilon S},$$

adding, and writing W for the heat per second H, we obtain :—

$$\theta_1 - \theta_4 = \frac{W}{S}\left\{\frac{1 + k_1 v}{k^2 v} + \frac{t}{k_3} + \frac{1}{\epsilon}\right\}.$$

We see, therefore, that the temperature rise at first diminishes with speed, and as the speed increases the first term in the bracket becomes constant, so that the temperature rise is independent of the speed of the machine.

The case of the enclosed motor is therefore on a

different footing to that of open or semi-enclosed machines.

It is seen from the above that inserting a fan inside the frame or casing has probably in the majority of cases but little effect. In a precisely similar way the equation could be worked out in the case where a fan blast acts on the outside of the case as a means of carrying off waste heat. The worst of such devices is the fact that the heat has to be transmitted from the working parts of the machine to the outside air, and in passing from the armature has to get through a layer of air.

In the above equation a conduction term might have been added to represent the heat carried off by foundations, etc.

The actual type of curve is shown in curve II in the above figure.

It appears from experiments to be referred to later that the temperature of the air in an enclosed motor is fairly uniform when the armature is rotating, the air being mixed or churned by the rotating armature fairly effectively.

It is well worth noticing that in cases where the temperature rise with time can be expressed by a formula such as that on p. 7 and which for brevity might be written

$$\theta = \frac{W}{S}(1 - \epsilon^{-\frac{S}{M}t}),$$

that if $$t = \frac{M}{S}, \text{ then}$$

$$\theta = \frac{W}{S} \cdot \frac{\epsilon - 1}{\epsilon}$$

or $$\theta = 0{\cdot}63\ \theta_{\max.}$$

The observation of such time is of considerable importance in cases where the machine is doing intermittent work, and has to be rated accordingly. The worst feature of attempting to discuss rating of machinery is the fact that it is the temperature rise of the insulating material which is the limiting factor. Meanwhile temperature of armature, commutator surface, mean temperature of armature by a resistance measurement are all used in referring to temperature rise. The temperature of the conductors, iron, insulation, and the air all differ from one another so that the temperature of a dynamo is difficult to specify.

Natural Cooling.

Natural cooling includes the loss of heat by convection, conduction and radiation.

In such cases as that of a hot body cooling by convection of air currents very little is known at present regarding the efficiency of such an arrangement. The very formidable nature of the problem presented is at once apparent when one considers that in a gas the pressure, density and velocity vary from point to point.

The general equation for the flow of heat in a liquid has been given by Fourier and Poisson, and is:—

$$\frac{d\theta}{dt} = \frac{k}{c}\left\{\frac{\partial^2\theta}{\partial x^2} + \frac{\partial^2\theta}{\partial y^2} + \frac{\partial^2\theta}{\partial z^2}\right\},$$

where θ is the temperature of the fluid at the point $x\ y\ z$, c is the capacity for heat, and k the conductivity. This equation, together with the ordinary hydrodynamical equations, and Euler's equations, enable solutions to be obtained for the cooling of certain bodies under certain

assumed conditions. Unfortunately these are not at all applicable to such a thing as, say, the cooling of a totally-enclosed motor by the convection of air.

The importance of these effects may be estimated from the following experiments. Mr. D. McFarlane made experiments on the loss of heat from blackened and polished copper balls about two centimetres radius in air. The value of the emissivity, or heat lost per second per square centimetre per degree difference of temperature, between the surface and the walls of the enclosure surrounding the ball, and the ball itself, was :—

$$\text{Black surface } e = 0{\cdot}000238 + 3{\cdot}6 \times 10^{-6}\,\theta^{\circ};$$
$$\text{Polished ,, } e = 0{\cdot}000168 + 1{\cdot}98 \times 10^{-6}\,\theta^{\circ}.$$

Dr. J. T. Bottomley tested the same two copper globes in vacuo, and found :—

Black surface $e = 0{\cdot}000125$ to $0{\cdot}000095$,
Silvered ,, $e = 0{\cdot}000035$,, $0{\cdot}000029$,

for different temperature ranges.

The coefficient of emissivity is itself a function of the temperature.

The relative importance of radiation and convection can, however, be well gauged from these experiments. From these experiments a black body loses in air per second, at the temperature of the experiments, about twice as much heat as it does in vacuo, when all convection was stopped. Of course the amount of heat lost by convection and radiation will depend on the form of the body. If the convection currents have free access to the various parts of the machine a great amount of heat may be carried away.

In order also to promote radiation a dark-coloured machine is preferable to one painted a lighter colour.

In all convective processes for getting rid of heat such as chimneys, circulation of water for heating purposes, the currents set up are due to gravity acting on the columns of differing density. The importance of convection arises from the fact (vide *Theory of Heat*, Clerk Maxwell, p. 250), viz. " the actual diffusion of heat from one part of the fluid to another takes place, of course, by conduction; but on account of the motion of the fluid the isothermal surfaces are so extended, and in some cases contorted, that their areas are greatly increased while the distances between them are diminished so that true conduction goes on much more rapidly than if the medium were at rest." Mere heating in itself is insufficient to cause convection currents, and without gravitation no such currents could exist. The expression "convection currents due to heating" is commonly used, however.

The ordinary theoretical expression for the velocity of the draught in a chimney

is
$$v = \sqrt{2gh \cdot \frac{T_1 - T_0}{T}},$$

where h is the height of the chimney, g is the gravitational acceleration, T_1, T_0 are the temperatures of hot and cold air respectively.

In this connection it is interesting to notice that theoretical values for the best ratio of internal to external temperatures of air have been deduced by Rankine (vide *Steam Engine and other Prime Movers*), and by using economizers a great saving in fuel is effected.

We see, therefore, that before convection currents can be obtained for cooling a certain amount of heat has to be dissipated, just as chimney draughts cost so much per annum for fuel.

When air ascends it cools and we have :—

$$-\frac{d\theta}{dx}=\frac{K-1}{K}\frac{T_0}{H_0},$$

where $H_0 = 7990$ metres, $T_0 = 273°$, $K = 1{\cdot}404$.

Hence $$-\frac{d\theta}{dx}=\frac{1}{101}.$$

This calculation is made on the supposition that the air is perfectly dry, and that it cools by expansion. Warmed air will not be able to ascend if the decrease of temperature upwards is less than above. Ordinary air cools, then, about 1° C. for every 100 metres, owing to rarefaction.

We see, therefore, that owing to the convection currents we should expect a drop in temperature between, say, the casing of a totally-enclosed motor and the air. In the chapter on heating and cooling curves some examples of this are given. The cooling surface of coils, etc., subjected to merely convective cooling, is based on allowing so many watts to be dissipated per square centimetre of surface. This is not correct, since the form of the body may modify greatly this action apart altogether from other considerations as to whether the body is placed vertically or horizontally. For magnet coils in open-type machines we have :—

$$\theta° = \frac{W}{A}\times 10.$$

Where $\theta°$ is rise of temperature, W is watts dissipated, and A is area in square decimetres.

These formulæ can only be determined by experiment for the different types of coils, etc., used.

In transformer oil cooling again the convective currents might be regarded in the above way.

Very often the transformer is merely immersed in oil, and no attempt is made to improve the circulation. The amount of oil required will depend, of course, on the rate of heating and the effectiveness of the cooling arrangements. If a transformer is oil cooled we must have, when it reaches its steady temperature rise above the iron case,

$$k_1 s M \theta_1{}^\circ = \text{Watts dissipated.}$$

This must equal $k_2\,S\theta_2$ where S is the surface of case and θ_2 difference between iron and air temperature.

In this k_1, k_2 are merely constants, s the specific heat of the oil used, M the mass of oil heated through θ° per second. Knowing, then, the watts to be dissipated, rise of temperature of the oil, the quantity per second can be estimated. Since the convection current velocity will depend on a great many factors, experimentally determined data is more satisfactory (see chapter on Heating and Cooling Curves).

As for radiation, we see from the McFarlane and Bottomley experiments that for a sphere in air it accounts for about half of the heat loss. The formula generally accepted for radiation is that of Stefan, viz.:—

$$S = \int_0^\infty E d\lambda = cT^4.$$

Where S is the total radiant energy, T the absolute temperature, and c is a constant which might be taken as about 124×10^{-10} for absolute temperature calculations.

If we write θ° for the difference of temperature between the body and the enclosure, then supposing this is small, we have:—

$$H = 4AkT^3\theta,$$

where H is the heat radiated per second, A is the area of surface, k is a constant.

It appears from experiments referred to at the beginning of this chapter that at ordinary temperatures, say 20° to 40° C., the radiation and convection are about equal, but at higher temperatures convection becomes more important.

As regards conduction the concrete would form an excellent sink for heat. Ideal cases of source and sink are easily worked out by Fourier's methods. We might assume for a dynamo working at constant output after all temperatures had become steady, that the temperature of the concrete is obtained from the usual expression $Q = -KA\frac{\partial\theta}{\partial x}dt$, which gives

$$\theta = \theta_0(1 - ax),$$

where θ_0 is the temperature of the surface, x the depth, a some constant, and θ the temperature at depth x.

Very little appears to be known regarding the quantity of heat conducted away by foundations, but it is seemingly a small quantity since measurements of heat carried off by the air passing through a generator appear to account for all the losses.

It seems to the author that in practical work more use should be made of the cooling curves of dynamos and transformers, since the temperature at any instant is fixed by the difference between the rates of heating and cooling.

The difficulty arises from the fact that the cooling curve should be taken with the armature replaced by a similarly shaped wooden one to create a draught, otherwise the curves taken from a dynamo with the armature at rest will probably differ greatly from the

cooling with the armature in motion. In the case of transformers with no moving parts, a study of the cooling curve is quite as important as studying the heating curve. By doing this, and making alterations in the methods of cooling, the effect of each alteration could be compared.

If curves of temperature rise with time are plotted, then the instantaneous temperature is settled by:—

$$\frac{d\theta_1}{dt} - \frac{d\theta_2}{dt},$$

where θ_1 and θ_2 are the heating and cooling temperatures at the same instant.

When $$\frac{d\theta_1}{dt} = \frac{d\theta_2}{dt}$$

the slopes of the curves or rates of change are identical, so that the machine is at a constant temperature.

Heating and Cooling Curves.

With regard to the tests discussed in the chapter on Ventilating Action of an Armature, a series of heat tests at light load were made of the same motor referred to running as open, semi-enclosed and totally-enclosed machine.

The heating and cooling curves are plotted in a time base in the following figure (p. 18). These curves all refer to the air temperature inside the motor case.

Another series of tests give the temperature at the surface of the case, a thermometer being arranged with its bulb immersed in mercury in a small copper calorimeter placed on the top of the case. These results,

heating and cooling, are shown in the figure on p. 19. The heating curve for the case of the open motor is practically identical with that for semi-enclosed, so that it is omitted in Fig. 5.

The temperature of the laboratory varied between 16° and 17° C. during these experiments.

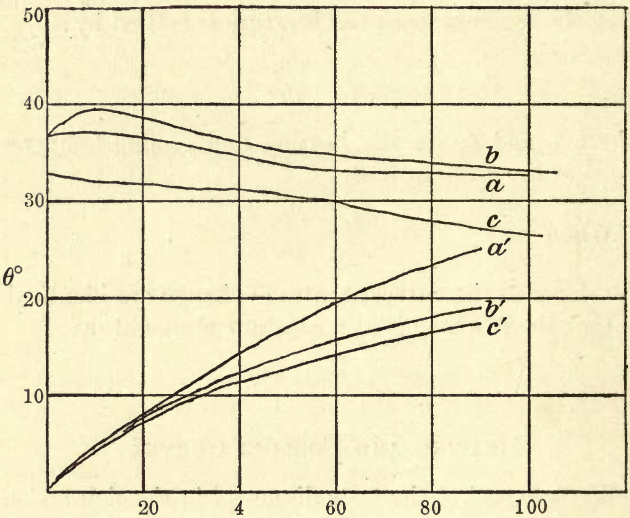

FIG. 4.—Time in minutes.

a a′ totally-enclosed motor cooling and heating.
b b′ semi- „ „ „ „
c c′ open motor.

It is of some importance as regards ventilation to compare the ratio of the temperature of, say, the magnet windings (deduced from a resistance measurement) with that of the air inside the case. Naturally this ratio is highest for the open type motor, and lowest for the semi-enclosed motor. In the latter case the ratio soon

appears to become quite constant. In the case of the totally-enclosed motor apparently the increase in temperature of the F.M. coils is greater than the increase in temperature of the air, so that the ratio lies between that of the open and semi-enclosed types. It is apparent, however, that by means of such curves the temperature of the air inside the motor case bears some ratio at any time to the temperature of the magnet copper temperature. In the case of the semi-enclosed

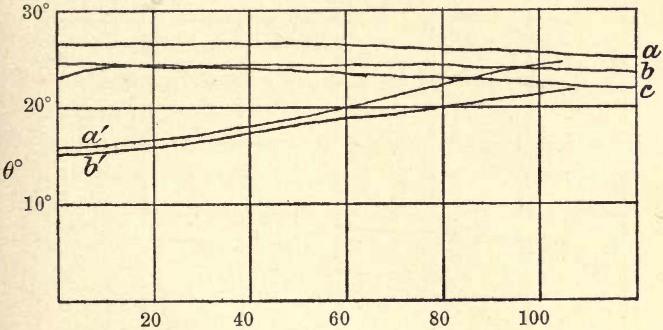

FIG. 5.—Time in minutes.

a a′ totally-enclosed motor cooling and heating.
b b′ semi- ,, ,, ,, ,,
c open motor.

motor this ratio is practically constant after ten minutes, in the open motor after about forty minutes, and in the totally-enclosed motor after about sixty minutes.

The manner in which heat flows out of a hot body is demonstrated in Fourier's *Analytical Theory of Heat*, where it is shown that the movement of heat must satisfy the equation:—

$$\frac{dv}{dt} = \frac{K}{CD}\left\{\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right\}.$$

If we consider a conductor surrounded by an insulating cylinder, the conductor being at some temperature θ_1, and if we suppose air moving with some velocity V at temperature θ_2 moved over the surface, the equation

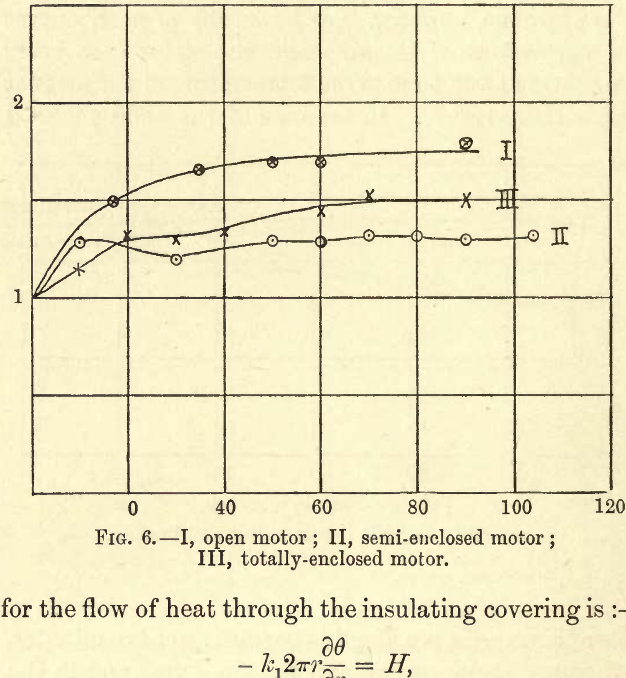

FIG. 6.—I, open motor; II, semi-enclosed motor; III, totally-enclosed motor.

for the flow of heat through the insulating covering is :—

$$-k_1 2\pi r \frac{\partial \theta}{\partial r} = H,$$

or $$\theta_1 - \theta_2 = \frac{H}{2\pi k_1} \log_\epsilon \frac{r_1}{r_2},$$

where H is the heat emitted per unit area per second, k_1 is the conductivity of the insulating material, r_1, r_2 the radii.

We see, therefore, that the difference in temperature will increase with H or watts dissipated.

It is shown (see "Convection of Heat from a Body cooled by a Stream of Fluid," A. Russell, M.A., D.Sc., *Physical Society Pcdgs.*, 1909) that

$$H = k\sqrt{V \cdot r_1},$$

where k is a constant, V is the velocity of the air, r_1 is the external radius. Substituting in the equation above we find that the ratio of temperatures is given by :—

$$\frac{\theta_1}{\theta_2} = 1 + K\sqrt{Vr_1}\ \log_\epsilon \frac{r_1}{r_2}.$$

The temperature of the air being assumed zero° centigrade.

It is easily shown by differentiation of the formula for temperature difference with regard to thickness of insulating material that

$$\frac{\partial \theta_1}{\partial T} = \frac{H}{k_1 r_1{}^{3/2}}\left\{r_1{}^{\frac{1}{2}} - \frac{k_2}{\sqrt{V}}\right\},$$

so that if r_2 is less than $\frac{k_2{}^2}{V}\ \frac{\partial \theta}{\partial T}$ is negative, so that a layer of insulation may actually cool a wire. This has been amply demonstrated by experiment. The letters k_1 and k_2 are constants.

The paper referred to above (by Dr. Russell) is most interesting, and should certainly be consulted by those engaged in designing electric generators, etc.

In the figure shown above where the ratio is plotted on a time base we might write :—

$$\frac{\theta_1}{\theta_2} = 1 + A(1 - \epsilon^{-\alpha t})$$

as a general equation. The limiting ratio of temperatures is :—

$$\frac{\theta_1}{\theta_2} = 1 + A = \text{a constant},$$

and the constants are easily determined and are of interest as showing how the heat generated is escaping.

With reference to various cooling media the following curve from Mr. R. D. Gifford's paper[1] is of interest as showing the proportionality of temperature of media to the temperature of the hot body.

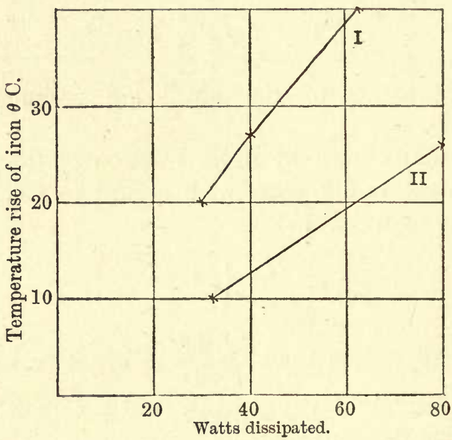

FIG. 7.—I, iron cooled by air blast; II, iron cooled by oil.

Another important figure (Fig. 8) is that showing the temperature gradient between the hot iron and cold case, with oil between them. This curve is of interest as indicating that the media is always at a lower temperature than the hot body, in this case nearly 15° C., and hotter than the cold outside case by about the same amount. Naturally one would

[1] "The Influence of various cooling Media upon the Rise of Temperature of Soft Iron Stampings," *Journal of the Institution of Electrical Engineers*, vol. 44, p. 753.

expect convection currents to keep the temperature of the oil throughout fairly uniform.

The effect of various different methods of ventilation is shown in Fig. 9, which has been plotted from the results given in the same paper.

These curves illustrate the tremendous difference between different methods of ventilation. The effectiveness of oil is clearly demonstrated, and in the case

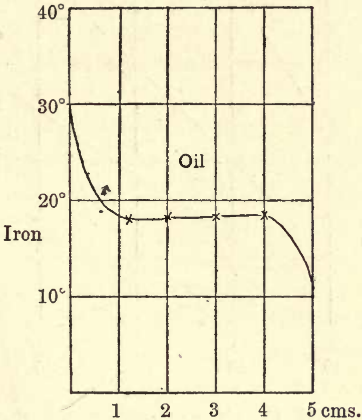

FIG. 8.—Temperature gradient.

where water pipes and oil are used, 90% of the heat generated is carried off by the water.

In calculating the quantity of air necessary to cool a given generator for a certain kilowatt waste, the temperature gradients through the insulating material must be carefully considered.

The author made some experiments on micanite, paxolin and other insulating material by passing steam at the same rate and at approximately 100° C. through tubes of approximately the same internal diameters but

different thicknesses. The external temperatures were measured by means of a thermometer, the bulb of which made contact with the middle of the tube and was covered with tinfoil.

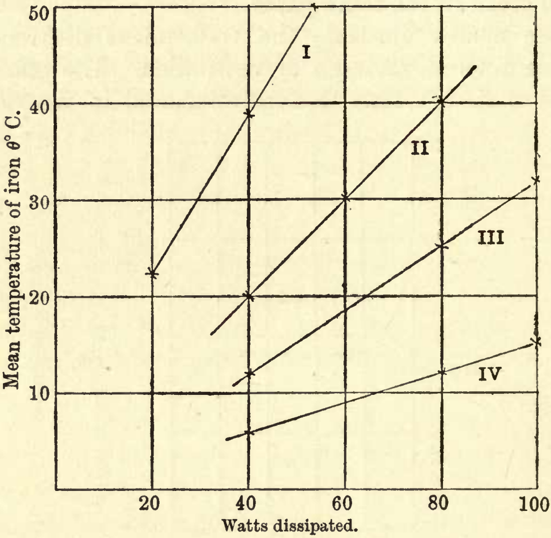

FIG. 9.—I, II, III, IV, air, air blast, oil, and oil and water worm respectively.

The results were as follows:—

TEST OF MICANITE TUBES.

No.	Ex. Diam. cms.	In. Diam. cms.	θ_1.	θ_2.	θ_1/θ_2.	Remarks.
1	1·22	0·62	100	47·7	2·1	Temperature of air 15·5° C.
2	1·16	0·59	100	51·5	1·94	
3	0·98	0·60	100	54	1·85	Tubes approximately horizontal.

A similar experiment was made with Paxolin Tubes.

No.	Ex. Diam. cms.	In. Diam. cms.	θ_1.	θ_2.	θ_1/θ_2.	Remarks.
1	0·94	0·61	100	50·5	1·98	Laboratory 16° C.
2	1·2	0·63	100	49	2·05	Tubes approximately
3	1·64	0·94	100	53	1·9	horizontal.

A tube of glass, 0·65 internal diameter, 0·88 cms. external diameter, tested in the same way gave with $\theta_1 = 100°$, $\theta_2 = 50·1°$. From this it would appear that the conductivity of micanite is about 1·55 times that of glass, so that if glass be taken as 0·0005, micanite is 0·000775.

The gradient actually existing in a generator between the copper conductors and the air outside will depend on the presence of the iron which conducts away the heat readily, and on the air draught as already mentioned.

Since substances like micanite, etc., soften at about 90° C., it is clear the copper conductors should not exceed this temperature, and the air outside cannot then be more than 45° C. in a well-ventilated machine.

Further experiments are being carried out on the temperature gradients actually existing under more practical conditions.

Ventilating Action of an Armature.

It has been already shown that there is a direct connection between the temperature rise of an armature and its speed. It remains to consider the best method of measuring the "ventilating effect" of an armature or rotor.

The problem presented by ventilation of dynamos is

essentially a complex hydrodynamical one. Except as general guides, the solutions obtained are hardly applicable to practical cases. Nevertheless there are one or two points to be borne in mind.

Any rotating mass sets air in motion, adds to its own inertia by doing so, and wastes energy in friction, etc. A revolving disc will cause a spiral flow to the periphery from the centre.

A cylinder revolving in still air (regarded as incompressible) sets the air in motion. If we regard the axis of the cylinder as the z axis, then we can consider the motion in the plane xy. The velocities are then—

$$u = -\omega y, \quad v = \omega x,$$

where ω is the angular velocity at the point.

The moment about the axis is :—

$$\mu r \frac{d\omega}{dr} 2\pi r \,.\, r$$

and since the angular velocity is a function of r, and the fluid is neither gaining nor losing angular momentum on account of the steady motion, we have :—

$$\omega = \frac{A}{r^2} + B$$

where r is the radius to the point, and A and B are constants. If the radius of the cylinder is a, and its angular velocity $\omega_\circ$ then :—

$$\omega = \omega_\circ . \frac{a^2}{r^2}$$

if the cylinder is revolving in an infinite fluid.

The couple due to friction on the cylinder varies as :—

$$a^2 \omega_\circ .$$

In the case of certain types of motors, such as poly-

phase induction motors and cylindrical rotors for alternating generators, we have a cylinder revolving inside a coaxial cylinder.

In this case if a is the radius of the rotor, and b the radius of the stator :—

$$\omega = \omega_{\circ} \cdot \frac{a^2}{r^2} \cdot \frac{b^2 - r^2}{b^2 - a^2}$$

we see that the angular velocity in the air gap is less than when the cylinder revolves freely in the air.

The frictional couple also increases greatly in this case and will depend for a given fluid on :—

$$a^2\omega_{\circ} \cdot \frac{b}{2\Delta r} \quad \text{approximately,}$$

Δr being the width of the gap. We see that in the case of a narrow gap the multiplication may be considerable. For any gap it is :—

$$a^2\omega_{\circ} \frac{1}{\left(1 - \frac{a^2}{b^2}\right)}$$

which approaches the previous value when b is great compared with a. Many interesting cases are dealt with in Lamb's *Hydrodynamics.*

The case of the coaxial cylinders with a narrow gap might be regarded as a first approximation to the case of an electrical machine.

If a fluid was forced along the surface, say, of the inner cylinder from one side a particle close to the surface would describe a helix in passing through the machine. Since for a helix

$$x = a \cos \theta, \; y = a \sin \theta, \; z = a\,\theta \,.\, \tan \alpha$$

where a is the radius of the helix, α the angle of the helix.

If V is the velocity of the cylindrical surface, then

$$\frac{dx}{dt} = -V \sin \theta, \ \frac{dy}{dt} = V \cos \theta, \ \frac{dz}{dt} = V \tan \alpha$$

and since $v = \sqrt{\left(\frac{dx}{dt}\right)^2 + \left(\frac{dy}{dt}\right)^2 + \left(\frac{dz}{dt}\right)^2}$

then $\qquad v = V \sqrt{1 + \tan^2 \alpha} = V \sec \alpha$

and $\qquad \tan \alpha = \frac{\frac{dz}{dt}}{V}$, where $\frac{dz}{dt}$

is the inflow velocity.

In this case we see that if the fluid entered cold it would rapidly heat up as it passed along the cylinder, becoming more useless for cooling purposes as its temperature rose. The higher the peripheral velocity of the rotor the smaller α would become and the longer the path over the armature surface.

An important equation for fluid flow is that of Bernouilli, viz. :—

$$z + \frac{p}{\omega} + \frac{v^2}{2g} = \text{constant};$$

z is the height above some datum level, p the pressure, ω the weight of unit quantity, v the velocity, and g gravitational acceleration. We see that in any stream lines if z is constant then as the pressure increases the velocity diminishes, so that :—

$$\frac{p_1}{\omega} + \frac{v_1^2}{2g} = \frac{p_2}{\omega} + \frac{v_2^2}{2g}.$$

When ventilating ducts are used the rotors will act more or less as a fan, cold air passing through the ducts. This is discussed more fully in a later chapter, viz. Practical Methods of Ventilating Armatures.

When, however, such an arrangement is used in a magnet frame inequalities of pressure are produced periodically. This is owing to the magnet poles: as the ducts pass from under them they are fully opened, and air can escape; when under them the air is practically imprisoned. The number of gusts per minute is:—

$$Nn\mathsf{N}$$

where N is the number of poles,
n „ „ revolutions,
N „ „ ventilating ducts.

The fact that the air when drawn in from both ends is practically baffled at the centre seems to the author to be a weak feature of this type of ventilation. In addition to this the radial slots form but a very inefficient fan since they are simply acting as a Barker's Mill, the air being the fluid instead of water in this case. Nevertheless engineers and designers of all kinds of electrical machinery seem to think this type of ventilation highly efficient and satisfactory. The efficiency of such a system of ventilation must be exceedingly low, that is to say,

the ratio, $\dfrac{\text{Power in air moved}}{\text{Power imparted to fan}}$ is probably about $\frac{1}{10}$th.

The "ventilating effect" of an armature might be measured by measuring the air drawn through the casing, and treating the armature as a fan. The efficiency of the fanning arrangement might be measured by running the armature for a short time with no ventilation, all the ducts being closed, and then noting the increased watts when the ducts and passages were opened. The volume of air moved might be measured so that "ventilating effect"

might be defined as efficiency as a fan, or "watts per cubic metre of air moved." This does not seem to offer any special difficulty except that referred to above, that in many machines the construction of the armature is such that ventilation is reduced to a minimum.

As a rule, when a machine is working there is a reduction of temperature due to the air supposed colder than the air inside the case of the machine being drawn into it. The author prefers, therefore, to use this reduction or increase of temperature due to starting or stopping an armature as a simple means of measuring its "ventilating effect" at the given speed, and temperature difference between air in machine and air in the space surrounding the dynamo or motor.

In any electrical machine one may assume that the heat generated is carried off by the natural diffusion of the molecules of the air, by eddying motion due to movement of its parts, by metallic conduction, and radiation.

In the case of a dynamo or motor one might regard the heat generated as being chiefly dissipated by the two former causes, and, following Prof. O. Reynolds,[1] write:—

$$H = A\theta + Bv\theta$$

where H is the heat carried off per unit surface per second, θ is the temperature difference between air and the surface, v is the velocity of air over the surface, and A and B are constants depending only on the state of the air.

In the case of a dynamo or motor the effect of the armature rotation is to induce a constant cooling effect

[1] "On the Extent and Action of the Heating Surface in Steam Boilers," *Pcdgs. Manchester Lit. and Phil. Soc.*, 1874.

so long as the velocity of rotation and temperature difference is constant. This effect corresponds to the second term above, and we see that the greater the difference of temperature between the surface and air the greater is the cooling effect for a given speed. In the case of a revolving armature, while it admittedly

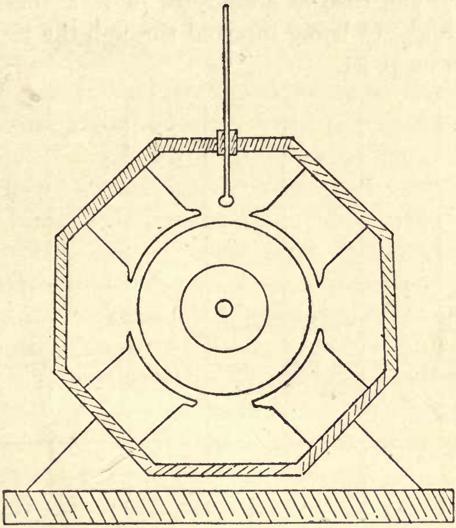

FIG. 10.—Diagram showing position of thermometer.

works as a fan it is also giving out heat, owing to its iron and copper losses.

The temperature, therefore, of a dynamo or motor will depend on heating due to armature, heating due to field coils, the natural cooling of the machine and cooling effect of armature rotation. If after running the machine the armature is suddenly stopped the temperature will depend upon the natural cooling of the machine.

The actual temperature of a dynamo or motor is a complicated question. The iron and copper in armature and fields all differ in temperature, and the air inside the case differs also in temperature from all of these. In order, however, to measure the ventilating action of any machine I use the temperature of the air inside the case as a measure of it, a thermometer reading to $\frac{1}{10}°$ C. being inserted through the top of yoke as shown on p. 31.

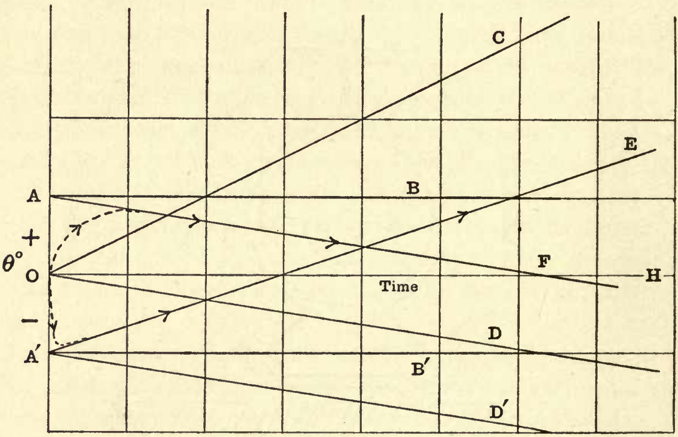

Fig. 11.—Hypothetical diagram.

Before discussing the results of experiment it is necessary to refer to the hypothetical diagram above.

Here O represents some temperature above that of the laboratory to which an imaginary dynamo or motor has been raised. Now suppose heat to be generated due to its losses and that this rate is constant; then let the line OC represent this. Again, suppose that the armature produces by its rotation a constant cooling effect, since temperature variation is now small, repre-

sented by the horizontal line parallel to OH at a distance below it equal to OA', and draw OD to represent the cooling not due to armature in the positions shown. If, then, we imagine the machine at a certain temperature, then on starting the armature running the temperature at any instant is given by the difference of the ordinates of the line OC and the line $A'D'$. Hence we obtain a series of lines forming our hypothetical curve, viz. OA', $A'E$.

Now suppose we reverse the process. Imagine the dynamo running at a certain temperature, the same as before, and we suddenly stop the armature; the result will be that by removing the cooling effect OA' we add this effect as a source of heating. Mark off $OA = OA'$. If the cooling effect is constant it is clear it may be represented by a horizontal straight line AB. And the lines OA, AF represent the instantaneous values of the temperature.

In practice, owing to the heat inertia of the machine, these lines are approximated to by the dotted curves shown in the figure, the armature when rotating being a more efficient cooler than a heater when stationary.

In order, then, to obtain the ventilating effect of an armature, all we have to do is to obtain at the given speed a diagram such as above, and by drawing tangents as shown; these give, where they cut the vertical axis, the ventilating effect due to the armature.

Hence the performance of any armature regarded as a fan can be readily determined and machines compared as regards ventilating properties at different temperatures or speeds.

Several experiments were made to determine this effect in the case of a motor working as open type semi-enclosed, and totally-enclosed, at practically

constant laboratory temperature and the same rise of temperature above laboratory in the case.

The results of the experiments and corresponding ventilating effects of the armature is shown in each case in the following diagrams.

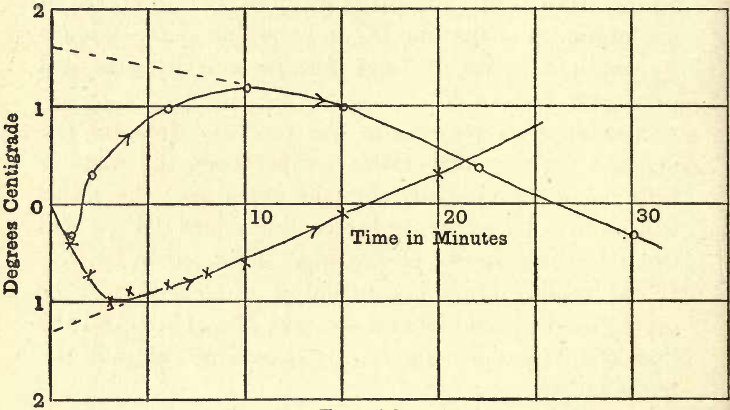

Fig. 12.—Test of Open Motor.

I.—Test of Open Type Motor.

Heating—Armature Revolving.				Cooling—Armature Stopped.			
Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.	Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.
0	17·2° C.	32·7°	—	0	17·2° C.	33°	—
1	,,	32·3	– 0·4°	1	,,	32·7	– 0·3°
2	,,	32	– 0·7	2	,,	33·3	+ 0·3
3	,,	31·7	– 1	6	,,	34	+ 1
4	,,	31·8	– 0·9	10	,,	34·2	+ 1·2
5	,,	31·8	– 0·9	15	,,	34	+ 1
6	,,	31·9	– 0·8	22	,,	33·4	+ 0·4
8	,,	32	– 0·7	30	,,	32·7	– 0·3
10	,,	32·1	– 0·6				
15	,,	32·6	– 0·1				
20	,,	33	+ 0·3				

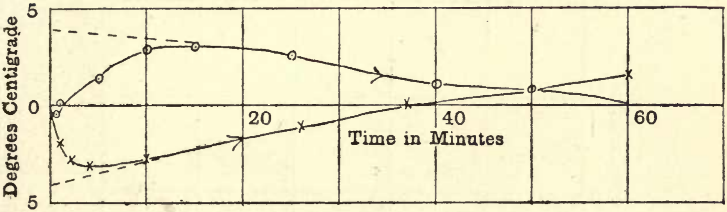

FIG. 13.—Test of Semi-enclosed Motor.

II.—TEST OF SEMI-ENCLOSED MOTOR.

Heating—Armature Revolving.				Cooling—Armature Stopped.			
Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.	Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.
0	17·6° C.	36·9°	—	0	17·1° C.	36°	—
1	,,	35	− 1·9°	0·5	,,	35·7	− 0·3°
2	,,	34·1	− 2·8	1	,,	36·1	+ 0·1
3	,,	33·7	− 3·1	5	,,	38·3	+ 2·3
4	,,	33·7	− 3·2	8	,,	38·9	+ 2·9
6	,,	33·8	− 3·1	10	,,	38·9	+ 2·9
8	,,	33·9	− 3	15	,,	39	+ 3
10	,,	34·1	− 2·8	25	17·5	38·6	+ 2·6
13	,,	34·5	− 2·4	40	,,	37·2	+ 1·2
16	,,	34·8	− 2·1	45	,,	36·9	+ 0·9
18	,,	35	− 1·9				
26	,,	35·8	− 1·1				
37	,,	36·9	0				
45	,,	37·6	+ 0·7				

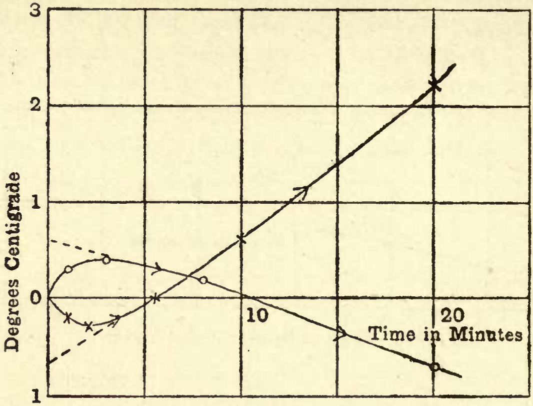

Fig. 14.—Test of Totally-enclosed Motor.

III.—Test of Totally-enclosed Motor.

Heating—Armature Revolving.				Cooling—Armature Stopped.			
Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.	Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.
0	17·2° C.	40°	—	0	18·4° C.	42·3°	—
1	,,	39·8	− 0 2°	1	,,	42·6	+ 0·3°
2	,,	39·7	− 0·3	3	,,	42·7	+ 0·4
4	,,	39·7	− 0·3	8	,,	42·5	+ 0·2
6	,,	40	0	20	,,	41·6	− 0·7
10	,,	40·6	+ 0·6				
20	,,	42·2	+ 2·2				

A series of tests were made during the process of raising the temperature of the motor to see whether the rise of temperature on stopping or drop on starting varied with the difference of temperature of the air in the motor case and the air in the laboratory, and the results are given in Fig. 15.

It appears, therefore, that for a given speed the above law is obeyed sufficiently for practical purposes.

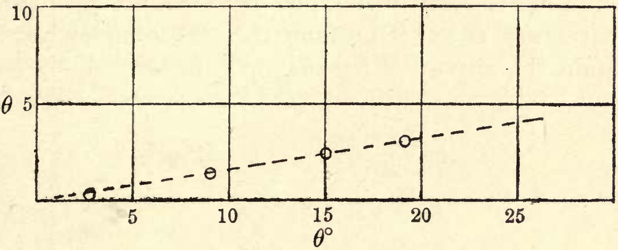

FIG. 15.

With regard now to the curves of temperature rise of any machine. If the temperature rise inside the case is used as a measure of the temperature rise of the machine, then the various ventilating effects may be separated.

This will be understood from the theoretical diagram (Fig. 16).

If there was no heat loss the temperature would rise with time along a straight line, so we might write

$$\theta = m_1 t;$$

but, as shown above, the cooling effect of the armature varies directly with θ for a given speed, so that if y is cooling effect, then

$$y = m_2\theta^\circ$$

hence $$y = m_1 m_2 t;$$

or $$\theta^\circ = kt.$$

It is plotted as a negative quantity.

We see, therefore, that curve $0B$ for the enclosed motor the armature cooling has little or no effect, $0B$ and the difference between the line $0A$ and curve $0B$

is simply due to cooling effect of the case. If we now deduct the armature effect we obtain the curve $0C$ for a semi-enclosed motor, and if we deduct again the ventilating effect of opening the case completely we obtain the curve $0D$ for the open motor.

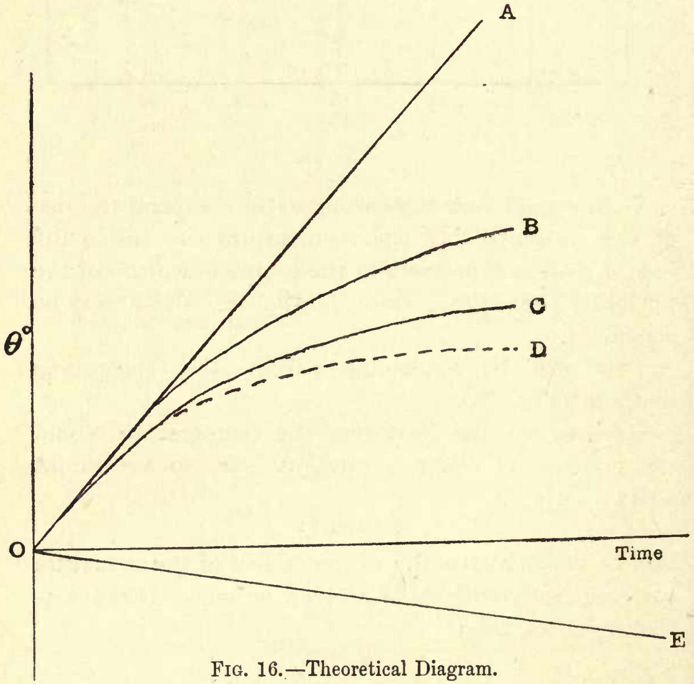

FIG. 16.—Theoretical Diagram.

The actual results obtained are shown in the following set of curves.

The line of total heating $0A$ is drawn as a tangent to the origin of the curve obtained from the motor running as totally-enclosed machine $0B$. The drop of temperature at different temperatures $0H$ is plotted to

the left of the ordinate; this deducted from $0A$ gives $0E$. The difference between the ordinates of $0A$ and $0E$ deducted from the totally-enclosed motor curve gives $0C$, the curve for the semi-enclosed motor; the difference between this curve and $0D$, the curve for the open motor, is the effect of opening the case.

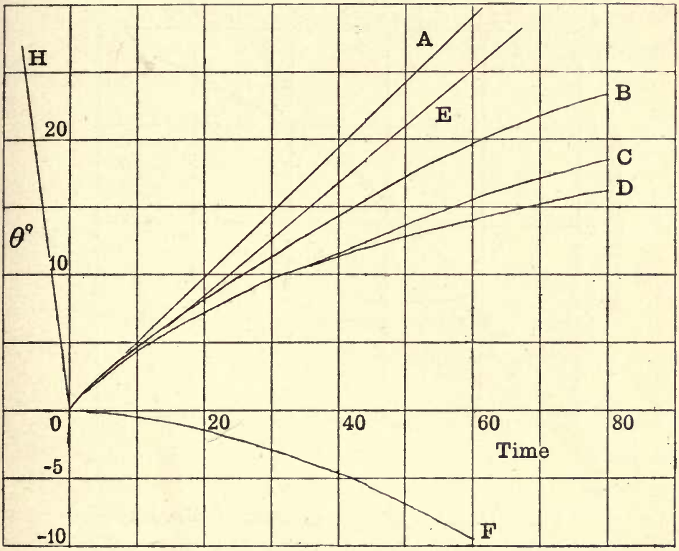

FIG. 17.—Actual Results.

Plotted negatively below the line is the cooling effect at different temperatures due to the case of the motor obtained by deducting the ordinates of $0B$ from $0A$.

It will be noticed that opening up the case has a relatively very great effect and semi-enclosing practically destroys this.

It must be borne in mind that in all the above tests the armature is giving out heat and so the cooling

effect due to the draught is not so large as it would be if it was rotating cold. The following curve was obtained by heating the semi-enclosed motor by magnet coils only, the armature remaining comparatively cool and then suddenly starting the armature. The cooling effect is noticeably much greater than in the previous experiments.

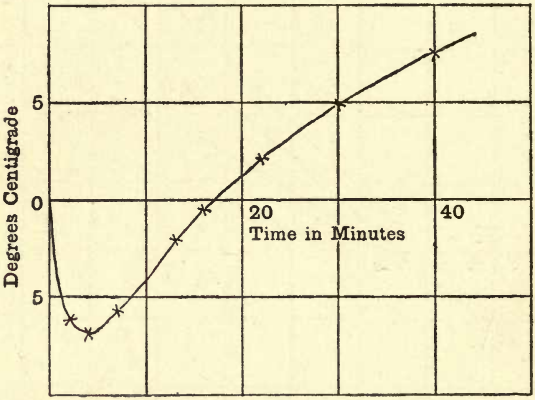

FIG. 18.

IV.—SPECIAL TEST. SEMI-ENCLOSED MOTOR.

Time in Minutes.	Laboratory Temperature.	Air in Case.	Variation of Temperature.
0	16·4° C.	36·1°	—
2	,,	29	− 7·1°
4	,,	28·2	− 7·9
7	,,	30·5	− 5·6
10	,,	32·9	− 3·2
13	,,	34·2	− 1·9
16	,,	35·7	− 0·4
22	,,	38·2	+ 2·1
30	,,	41	+ 4·9
40	,,	43·6	+ 7·5

It appears from the above experiments that an armature rotating inside a case which is totally enclosed is often practically devoid of any ventilating effect. After an enclosed motor has run for some time the temperature of the air inside the case is practically the same throughout. If there is some cold air the ratio of the mass of it to the rest is easily found. The temperature at the top of the case being t, that at bottom t_2, and the resulting temperature when the armature rotated again t_3, we have

$$m_1t_1 + m_2t_2 = (m_1 + m_2)t_3$$
$$m_1(t_1 - t_3) = m_2(t_3 - t_2)$$
$$\frac{m_1}{m_2} = \frac{t_3 - t_2}{t_1 - t_3}$$

Substituting $t_1 - t_3 = 0{\cdot}5$, $t_3 - t_2 = 7{\cdot}5$ obtained from an experiment,

we have $$\frac{m_1}{m_2} = \frac{15}{1} \text{ approximately,}$$

or, roughly speaking, the cold air is only 6·6 % of the hot air. Putting a fan on the shaft of a totally-enclosed motor can only mix the hot and cold air and can only be effective for ventilating purposes if there is oil circulation through pipes or some means of setting up a difference to temperature of the air inside, and outside the case.

Figs. 19 and 20 show elevation and section and principal dimensions of the motor used in the experiments referred to in the paper.

The motor was shunt wound 7 B.H.P. 400/1000 R.P.M. 480 volts. The cross section of yoke is 18¾ sq. in., pole pieces 6½ in. axially and 4 in. circumference.

Field Coils wound with 28 lb. 0·028 in. diam. wire each containing 4876 turns of wire.

The Armature Case is $6\frac{3}{4}$ in. long, including two end plates $\frac{1}{8}$ in. thick and one vent plate $\frac{3}{8}$ in. thick, the remaining stampings being 0·015 in. thick. It has 41 slots, 123 coils and 123 commutator bars 3 coils per slot.

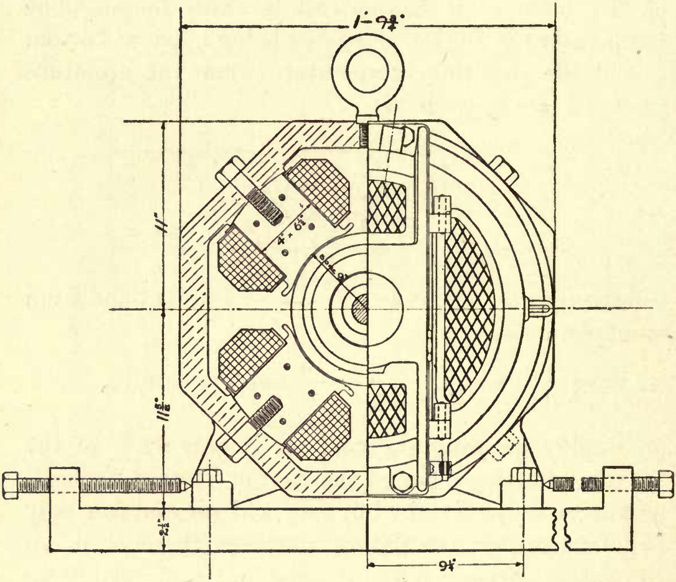

Fig. 19.—Semi-enclosed Motor used in Experiments.

Each coil consists of 3 turns, and each turn consists of 2 wires in parallel. The armature is wave round.

Temperature Rise Tests. The experiments referred to were all carried out at light load, and the inputs were practically the same in each case, viz. 1·9 amperes at 470 volts at 1000 revolutions. The mean shunt current was about 0·7 ampere.

A Kapp and Housmann test for eddies, hysteresis

and friction loss gave friction loss of 0·22 watt per revolution, hysteresis and eddies 0·34 watt per revolution at 1000 revolutions per minute. The C^2r loss in armature was negligible at light load, since the resistance was 0·9 ohm only.

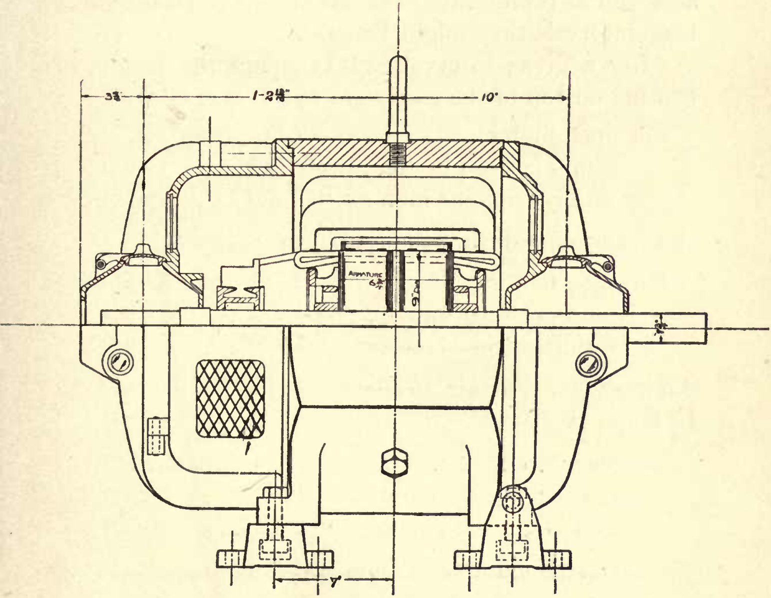

FIG. 20.

The ammeters used were calibrated after the test by comparison with Kelvin's balance and found accurate. The thermometers were also carefully tested and found accurate to $\frac{1}{10}$° C.

A series of experiments were also made on the heating and cooling of the top of the motor case by means of a thermometer inserted in a copper vessel containing mercury in contact with the case.

As a rule these experiments merely confirm the opinion of the author that mathematical expressions based upon assumptions of laws of cooling which take no account of the changed conditions when stopping or starting the armature require to be used with caution, although as representing the average performance over long intervals they might be used.

After a given interval, say $1\frac{1}{2}$ hours, the results of heating on top of the case were as follows :—

For open motor . .	+ 6° C.	from 14·7° C.
„ semi-enclosed motor .	+ 6° C.	„ 14·8° C.
„ totally-enclosed motor	+ 7·6° C.	„ 15·7° C.

and for cooling during the same time :—

For open motor . .	− 1° C.	from 24·5° C.
„ semi-enclosed motor .	+ 1·7° C.	„ 22·9° C.
„ totally-enclosed motor	+ 1·8° C.	„ 26·2° C.

Similarly for the air inside case the cooling in the $1\frac{1}{2}$ hours was :—

For open motor . .	− 5·7° C.	from 32·7° C.
„ semi-enclosed motor .	− 2·9° C.	„ 36·8° C.
„ totally-enclosed motor	− 2·8° C.	„ 36·5° C.

The mean laboratory temperature being about 16° C. in each case.

In making the experiments the case of the semi-enclosed motor was opened up when working as open motor, and enclosed by pasting paper over all openings and grids to make it a totally-enclosed motor.

In cases of breakdown of insulating material in field coils one would expect this to occur more frequently at the top of the motor case in the case of totally- or semi-enclosed motors, and after the armature cooling effect is withdrawn.

Since writing the above paragraph several instances of this have been brought to the author's notice. In one case a dynamo which was overloaded went on fire on stopping, to the surprise of those in charge.

THE VENTILATING ACTION OF FANS.

WHEN air flows over a surface which is at some higher temperature, there is an interchange of heat between it and the surface, depending on the difference of temperature between them.

If we assume a Newtonian law of cooling we would have:—

$$-\frac{d\theta}{dt} = A\theta^\circ$$

where A is a constant. Hence if θ_1 is the temperature of the air and θ_2 the original temperature of the surface the temperature after a time t is

$$\theta^\circ = \theta_1{}^\circ\left\{1 + \epsilon^{-At}\right\},$$

only after an infinite time would the temperature approach that of the stream of air.

In the case of the ventilation of dynamo machinery, the air necessary to cool the generator has to pass through more or less restricted channels, and, as is well known from Professor Osborne Reynolds' experiment, you may increase the pressure of the air indefinitely without increasing the temperature of the air very greatly. This is shown in Fig. 21 on next page, where θ is the increase in temperature of the air passed through a pipe of brass, or copper, warmed by boiling water.

In the case of a fan, the work done per minute is PQ, where P is the pressure, and Q the quantity of air passing per minute. Hence the horse-power $= \frac{PQ}{33{,}000}$.

Now the pressure is generally measured by a manometer, or Pitot tube, and is given by :—

$$\text{Velocity of air} = 4\sqrt{h}$$

where the velocity is measured in metres per second at

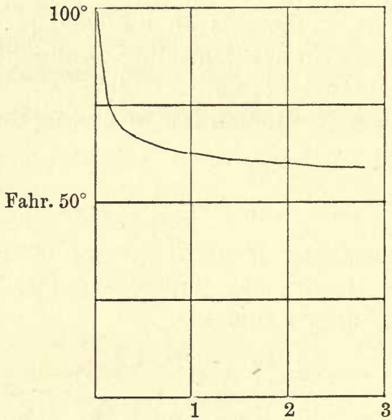

FIG. 21.—Pressure in inches of Mercury.

0° C., and h is the head in millimetres of water. The pressure, therefore, varies as v^2. Also $Q = Av$ where A is the cross section of the blast pipe somewhere and v the velocity. We might therefore write the power of the fan as :—

$$\text{H.P.} = kv^3$$

where k is a constant. We see, therefore, that if a fan is pumping air through a generator, if we double the velocity of the air we require eight times the power.

If we regard the heat carried off by a fan as due to the term

$$Bv\theta$$

in the Reynolds' equation, we have

$$\theta = \frac{1}{v}$$

and hence horse-power of fan :—

$$\frac{k}{\theta^3}.$$

If air is passed over a surface, then, there is a certain temperature difference which gives maximum economy of fanning. If we assume that v and θ vary inversely, as they appear to do, then the quantity of air required is

$$Q = \frac{Ak_1}{\theta}$$

where k_1 is a constant, A the effective area of ventilating ducts. Further the pressure of a fan blast may be written :—

$$k_2 v^2 \text{ or } P = \frac{k_1 k_2}{\theta^2}.$$

The power of a fan being measured by PQ, we have the power

$$P = pQ = \frac{K}{\theta^3}$$

where K is a constant.

We might assume for any machine ventilated with a constant air blast that

$$\text{Waste kilowatts} = S\theta$$

where S is some effective surface and θ the temperature difference between surface and air. Hence our equation stands :—

$$S\theta + \frac{K}{\theta^3} \text{ to be a minimum.}$$

Differentiating, we have

$$S - \frac{3K}{\theta^4} = 0$$

or

$$\theta = \left(\frac{3K}{S}\right)^{\frac{1}{4}}$$

for temperature difference of maximum economy.

That this is a minimum is shown by the fact that $\frac{d^2W}{d\theta^2}$ is positive.

Probably the simplest method is to solve each case graphically.

This question does not appear to have attracted the attention it deserves. As an illustration, I take the case of three transformers erected in a large central station, cooled by air blast, and it is required to find the temperature rise of maximum economy and input to fan.

TEST OF THREE SINGLE-PHASE STATIC TRANSFORMERS.

Primary volts 8000, secondary volts 180 ; 50 cycles per second.
Total output as 3-Phase 1650 kilowatts.

Load.	C^2R.	Iron Loss.	θ° rise.	Killowatts taken by Fan.
Full	14·8	15	35° C.	
$1\frac{1}{4}$	22	,,	50	6·75
$1\frac{1}{2}$	32	,,	65	

If W_i and W_c are iron and copper losses respectively

$$W_i + W_c = S\theta.$$

This gives a value of S of about 0·8 and $K = 6·75 \times 35^3$.

We find therefore :—

θ	$S\theta$	$\frac{K}{\theta^3}$
0	0	∞
10	8	287·3
17·5	14	53 6
35	28	6·7
70	56	0·84

Plotting these values, we obtain the curves below, showing that the temperature rise of maximum economy

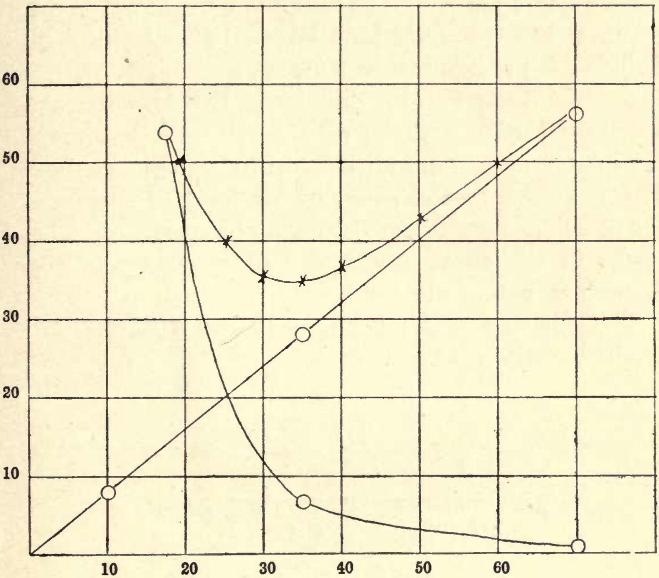

FIG. 22.

$S\theta$ and $\frac{K}{\theta^3}$ plotted vertically ; θ plotted horizontally.

is 32° C. approximately and input to fan 6·75 kilowatts. The transformer together with fan and motor are working at nearly their maximum economy at full load.

As regards the power necessary to carry off the waste heat by means of fans, the data at present appears to be very scanty. In the case of the transformer set just referred to, the fan-power is $\frac{37}{6\cdot75}$ at $1\frac{1}{4}$ load, or 5·5 kilowatts per kilowatt of fan-power. Again, in the *Electrician* of Nov. 27, 1903, there is an article on "The Testing of Electric Generators by Air Calorimetry," by Professor R. Threlfall, M.A., F.R.S., and it would appear that to carry off about 40 kilowatts required 5 kilowatts of fan-power, although less might have been required had the ducts been larger. This gives, then, 8 kilowatts per kilowatt of fan-power.

As the temperature difference between, say, an armature surface and the air increases, the fan-power per kilowatt waste diminishes rapidly for dynamos, etc. This is limited by the power of the insulation to resist the deteriorating effect of the higher temperature. This seems to the author to be one of the reasons why fanning is not of greater utility.

The efficiency of a machine with and without a fan might be written as previously :—

$$\epsilon_1 = \frac{O}{O + L + C^2R}$$

$$\epsilon_2 = \frac{kO}{kO + L + k^2C^2R + \dfrac{L + k^2C^2R}{p}},$$

p being the number by which the waste kilowatts have to be divided to give the power expended in the fan :—

$$\therefore \quad \epsilon_2 = \frac{O}{O + \dfrac{L}{k} + kC^2R + \dfrac{L + k^2C^2R}{kp}}.$$

Therefore $\epsilon_1 > \epsilon_2$, if,

$$L + C^2R < \frac{L}{k} + kC^2R + \frac{L + k^2CR}{kp}.$$

If
$$L + C^2R = \frac{L}{k} + k^2C^2R + \frac{L + k^2C^2R}{kp},$$

then
$$k^2 - \frac{p(L + C^2R)}{(p+1)C^2R}k + \frac{L}{C^2R} = 0,$$

or the values giving equal efficiency are :—

$$k = \frac{p}{2(p+1)}\frac{(L + C^2R)}{C^2R} \pm \frac{1}{2}\sqrt{\left(\frac{p}{p+1}\right)^2\left(\frac{L + C^2R}{C^2R}\right)^2 - \frac{4L}{C^2R}}.$$

From the quadratic, or by differentiation of the equation for k, we see that the efficiency will be a maximum when

$$k = \frac{p(L + C^2R)}{2(p+1)C^2R}.$$

The ratio giving k, then, for overrunning the generator is diminished in the ratio $\frac{1}{1 + \frac{1}{p}}$, and we see that if p is large, or the fan-power a small fraction of the wasted power, this will not differ greatly from the expression found above, in Chapter I.

It may as well be pointed out that exactly the same expression would apply in the case of a refrigerator, to be discussed later.

So far as the effect on the efficiency is concerned, the fan is responsible for the term :—

$$\frac{L + k^2C^2R}{kp}.$$

The following tables illustrate the increase of horse-power of fans with increase of quantity of air at constant pressure, and also increase of horse-power with increase of pressure, the quantity of air remaining constant.

"KEITH" CENTRIFUGAL STEEL PLATE FANS.
2-inch Water Gauge.

Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.	Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.
	in.				in.		
400	5	3350	0·24	16000	25	560	8·5
600	5	3400	0·36		30	470	7·2
800	5	3600	0·61	18000	25	580	10·3
	7½	1940	0·47		30	470	8·5
1000	7½	1940	0·56		35	400	8·0
1200	7½	1940	0·67	20000	25	600	12·2
1400	7½	1940	0·8		30	470	9·9
1600	7½	1950	1·0		35	400	8·9
	10	1440	0·9	25000	30	480	14·0
1800	7½	2040	1·3		35	400	12·0
	10	1440	1·0		40	350	11·2
2000	10	1440	1·1	30000	30	500	18·6
2500	10	1450	1·42		35	400	15·5
	12½	1150	1·37		40	350	13·7
3000	10	1490	1·9		45	310	13·2
	12½	1150	1·63	35000	35	410	19·8
	15	920	1·55		40	350	17·4
3500	12½	1150	1·96		45	310	15·6
	15	920	1·7	40000	35	430	24·5
4000	12½	1150	2·24		40	350	21·0
	15	920	1·85		45	310	18·7
4500	12½	1170	2·6		50	280	17·7
	15	920	2·1	50000	40	360	29·8
5000	12½	1200	3·2		45	310	26·2
	15	920	2·4		50	280	23·7
	17½	790	2·2		55	255	22·0
6000	15	920	3·2	60000	45	320	34·6
	17½	790	2·75		50	280	30·8
7000	15	960	4·2		55	255	28·4
	17½	790	3·5		60	235	26·6
	20	690	3·1	70000	45	340	44·5
8000	17½	810	4·3		50	290	39·0
	20	690	3·7		55	255	35·4
9000	17½	830	5·1		60	235	32·5
	20	690	4·5	80000	50	300	48·5
	25	550	4·0		55	260	43·6
10000	17½	860	6·0		60	235	40·0
	20	700	5·2	90000	55	265	52·0
	25	550	4·5		60	235	47·5
12000	20	730	6·9	100000	55	270	64·0
	25	550	5·5		60	240	56·0
14000	25	550	7·0	110000	60	245	65·0
	30	470	6·2	120000	60	250	74·0

"Keith" Centrifugal Steel Plate Fans.
3-inch Water Gauge.

Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.	Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.
	in.				in.		
500	5	4100	0·43	18000	25	690	13·8
600	5	4100	0·5		30	570	12·0
800	5	4200	0·7	20000	25	700	16·2
1000	5	4500	1·1		30	570	13·5
	7½	2400	0·86	25000	25	740	22·6
1200	7½	2400	0·99		30	570	18·8
1400	7½	2400	1·16		35	490	16·7
1600	7½	2400	1·34	30000	30	590	25·0
1800	7½	2400	1·57		35	490	21·2
	10	1780	1·53		40	430	19·8
2000	7½	2420	1·91	35000	30	600	31·5
	10	1780	1·68		35	490	26·5
2500	10	1780	2·1		40	430	23·7
3000	10	1800	2·6	40000	35	500	33·0
	12½	1410	2·5		40	430	28·6
3500	10	1830	3·2		45	380	26·5
	12½	1410	2·8	45000	35	510	39·2
4000	12½	1410	3·3		40	430	34·0
	15	1100	2·9		45	380	30·2
4500	12½	1410	3·7	50000	35	525	46·4
	15	1100	3·1		40	435	40·0
5000	12½	1410	4·2		45	380	35·5
	15	1100	3·4		50	340	33·0
6000	12½	1470	5·7	60000	40	450	53·0
	15	1100	4·2		45	380	46·3
	17½	970	3·9		50	340	42·0
7000	15	1100	5·4		55	310	40·0
	17½	970	4·7	70000	45	392	58·2
8000	15	1140	6·7		50	340	52·5
	17½	970	5·7		55	310	48·0
	20	850	5·3		60	290	46·0
9000	17½	980	6·8	80000	45	405	72·8
	20	850	6·0		50	345	64·5
10000	17½	1000	8·0		55	310	59·0
	20	850	7·0		60	290	54·0
	25	690	6·7	90000	50	355	77·5
12000	17½	1040	10·7		55	310	69·0
	20	860	9·2		60	290	65·0
	25	690	8·0	100000	50	365	91·2
14000	20	885	11·7		55	315	82·0
	25	690	9·6		60	290	75·0
16000	20	930	14·5	120000	55	330	109·0
	25	690	11·6		60	292	100·0
	30	570	10·6	140000	60	300	126·0

"KEITH" CENTRIFUGAL STEEL PLATE FANS.
4-inch Water Gauge.

Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.
	in.		
600	5	4700	0·7
800	5	4760	0·95
1000	5	5000	1·3
	7½	2740	1·2
1200	7½	2740	1·4
1400	7½	2740	1·6
1600	7½	2740	1·8
1800	7½	2740	2·0
2000	7½	2740	2·3
	10	2000	2·2
2500	10	2000	2·7
3000	10	2000	3·3
	12½	1600	3·4
3500	10	2040	3·9
	12½	1600	3·8
4000	10	2100	4·6
	12½	1600	4·4
	15	1300	4·1
4500	12½	1600	5·0
	15	1300	4·4
5000	12½	1600	5·5
	15	1300	4·7
6000	12½	1620	6·8
	15	1300	5·5
7000	15	1300	6·6
	17½	1110	6·2
8000	15	1300	8·2
	17½	1110	7·1
10000	17½	1120	9·9
	20	970	9·0
12000	17½	1160	13·1
	20	970	11·6
	25	790	10·7
14000	20	990	14·4
	25	790	12·4
16000	20	1020	17·9
	25	790	14·5
18000	20	1060	21·2
	25	790	17·2
	30	660	16·0

Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.
	in.		
20000	25	790	20·0
	30	660	17·6
25000	25	820	28·0
	30	660	23·4
	35	570	22·2
30000	30	660	30·6
	35	570	27·0
35000	30	670	38·8
	35	570	33·0
	40	500	30·0
40000	30	690	48·0
	35	570	40·0
	40	500	36·0
45000	35	570	49·0
	40	500	42·5
	45	440	40·0
50000	35	580	57·0
	40	500	50·0
	45	440	45·0
60000	40	500	66·0
	45	440	58·0
	50	400	53·0
70000	45	440	73·0
	50	400	66·0
	55	360	62·0
80000	45	450	89·0
	50	400	80·0
	55	360	73·0
90000	50	400	95·0
	55	360	87·0
	60	330	80·0
100000	50	410	113·0
	55	360	102·0
	60	330	94·0
120000	55	370	134·0
	60	330	122·0
140000	60	340	15·50
150000	60	344	17.20

"KEITH" CENTRIFUGAL STEEL PLATE FANS.
5-inch Water Gauge.

Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.	Cubic Feet per Minute.	Size of Fan Outlet Dia.	Revs. per Minute.	Horse Power.
	in.				in.		
600	5	5300	0·9	26000	25	900	35·4
800	5	5300	1·1		30	740	29·5
1000	5	5500	1·5	28000	30	740	32·8
	7½	3100	1·5		35	630	30·7
1200	7½	3100	1·7	30000	30	740	36·2
1400	7½	3100	1·9		35	630	33·0
1600	7½	3100	2·2	35000	30	740	45·5
1800	7½	3100	2·5		35	630	39·5
2000	7½	3100	2·8	40000	30	760	56·0
	10	2240	2·8		35	630	48·0
2500	10	2240	3·3		40	550	44·0
3000	10	2240	4·1	45000	35	630	56·5
3500	10	2240	4·9		40	550	51·0
	12½	1800	4·9	50000	35	640	67·0
4000	10	2270	5·8		40	550	59·0
	12½	1800	5·4		45	490	55·0
4500	12½	1800	6·0	55000	35	650	77·0
	15	1450	6·0		40	550	68·0
5000	12½	1800	6·7		45	490	61·0
	15	1450	6·2	60000	40	550	76·0
6000	12½	1800	8·3		45	490	68·0
	15	1450	6·9		50	440	66·0
7000	15	1450	7·9	65000	40	560	87·0
	17½	1260	7·8		45	490	77·0
8000	15	1450	9·5		50	440	72·0
	17½	1260	8·7	70000	45	490	87·0
10000	17½	1260	11·7		50	440	79·0
	20	1080	11·0		55	400	77·0
12000	17½	1270	15·6	80000	45	495	105·0
	20	1080	13·4		50	440	95·0
14000	20	1090	16·9		55	400	89·0
	25	890	15·5	90000	50	440	114·0
16000	20	1110	20·7		55	400	103·0
	25	890	17·7		60	370	100·0
18000	20	1140	25·0	100000	50	445	132·0
	25	890	20·5		55	400	121·0
20000	25	890	23·9		60	370	112·0
	30	740	22·2	110000	60	370	127·0
22000	25	890	27·4	120000	55	405	158·0
	30	740	24·2		60	370	145·0
24000	25	895	31·0	140000	60	370	182·0
	30	740	26·7	150000	60	375	204·0

The worst feature of fan ventilation seems to be that the pressure necessary to send the air through the windings varies greatly according to the design of the machine. In some cases $\frac{1}{2}$ in. head of water is sufficient to drive the required quantity of air through the machine, whereas in other cases $2\frac{1}{2}$ or 3 ins. may be necessary.

With increasing losses in the machine, the fan-power necessary to carry off the heat increases enormously. The orifices remaining the same, in order to double the quantity of air would require about eight times the power previously used.

It depends also on atmospheric conditions, and in warm climates the output of generators is seriously diminished owing to high shade temperatures of air. Dust and grit are also mentioned as objections.

It would appear that the fan is destined to play only a secondary part in the cooling of very large generators in the future, but at present smaller generators should undoubtedly take full advantage of fanning.

The great saving of power which results in fanning is realized in drying installations, where the difference of temperatures are kept as great as possible.

In all cases, whether a fan is used to pump air through the machine, or a pump in the case of oil-cooled transformers, the condition to give maximum economy ought to be considered.

At present this is certainly not done, and many motors fitted with fans which the author has examined would probably work better without them. This state of matters is, however, passing away.

No doubt in this country the temperature of the air outside an engine-room is considerably cooler than the generator or transformer. At the same time there is a

considerable difference of temperature, probably 20° to 30° C., necessary to get the heat to pass readily from the windings to the ventilating air current. This is clearly shown in Fig. 21 above. The same thing is again evident in oil cooling, although in the case of oil the difference is not nearly so great as with air (see Fig. 8, p. 23.

There is, of course, no comparison between air and oil cooling for such things as static transformers. It is all the more difficult to understand why air cooling for such devices is adopted at all. Besides being less effective, the risk of breakdown is enormously increased, since with oil cooling, owing to the mass of the oil and its higher specific heat, even if the circulation was stopped, some time would have to elapse until the temperature rose to a dangerous extent.

The Refrigerator as a Cooling Agent.

It seems somewhat remarkable that the only systems used to cool generators and transformers are air and oil cooling, or a combination of air, oil and water cooling.

In the meantime many large generators are almost entirely fan cooled, the air entering through ducts in the base-plate, generally beneath, and escaping at the top. In certain cases water-pipes have been inserted in the air-ducts, thereby cooling the air before it reaches the machine.

This, of course, is all very well in temperate climates, where cool shade temperatures or cold water are to be had easily. There are, however, many cases where

these are not available, and the output of the machine is seriously limited by the high initial temperature. Further, in the case of heating due to overloads, a smaller generator might suffice, provided the excessive rise of temperature was removed.

The most economical method of generating cold is undoubtedly by means of refrigerators, provided the temperature range is not too great.

Thermodynamically we might write

$$h = H \frac{\Delta T}{T}$$

where h is the heat necessary to neutralize a larger quantity of heat H, through a range ΔT, of working temperature and absolute temperature T.

If we differentiate this expression with regard to time, $\frac{\Delta T}{T}$ being a constant, we have :—

$$\frac{\partial h}{\partial t} = \frac{\partial H}{\partial t} \frac{\Delta T}{T}.$$

Or if we express it in electrical units,

$$w = W . \frac{\Delta T}{T}$$

where w is the number of kilowatts required to neutralize a greater number of kilowatts W, through the range of temperature ΔT.

For instance, suppose 100 kilowatts were to be neutralized through 10° C., the absolute temperature being 300° C., then

$$w = 100 \times \frac{10}{300} = 3{\cdot}\dot{3} \text{ kilowatts.}$$

Of course, the thermodynamic limit is not reached in ordinary refrigerators, but for a 10° C. range of

temperature we have the following coefficients of performance :—

Type of Machine.	Coefficient of Performance.
Ammonia . .	27·8
Carbonic Acid . .	21 } Upper limit of temperature 20° C.
Theoretical . .	28·3

(See Ewing's *Mechanical Production of Cold.*)

So far as the actual performance is concerned, the ammonia and carbonic acid machines approach very closely the maximum limit.

It remains now to consider the question of temperature rise. If W is the total kilowatt loss in, say, the stator windings of a generator, R_f the kilowatts neutralized by means of refrigeration, S the surfaces cooled, M the total mass heated, and s the specific heat, then:—

$$sM\frac{d\theta}{dt} = W - R_f - S\theta$$

$$\frac{d\theta}{dt} + \frac{S}{sM}\theta = \frac{W - R_f}{sM}.$$

If the windings are at their maximum temperature, $\frac{d\theta}{dt} = 0$, and we have

$$\theta^\circ = \frac{W - R_f}{S}.$$

It is clear that if $W = R$, the temperature rise $\theta^\circ = 0$, and the windings are at atmospheric temperature.

If, then, by circulation of oil or air we had a refrigerator to neutralize the stator loss, a great increase in output might be obtainable. If W is the kilowatt loss at full load, R the refrigerated kilowatts, then, if

we assume the output proportional to the square root of the losses, we would have—

$$\frac{O_2}{O_1} = \sqrt{\frac{W}{W - R_f}}.$$

Now, if R_f is only a small fraction of the losses, we see that, approximately,

$$\frac{O_2}{O_1} = 1 + \tfrac{1}{2}\frac{R_f}{W};$$

and this shows that so far as output is concerned a small amount of cooling is not very effective.

In most generators the stator windings are inserted in micanite tubes, and the temperature gradient as already discussed might be assumed as

$$\theta_1 - \theta_2 = kW \log_\epsilon \frac{r_2}{r_1}$$

where W = kilowatts dissipated, $r_2 r_1$ the external and internal radii, and k is some constant.

Now, if θ_1 is the internal temperature of the tube, and θ_2 the external temperature, then, if by some means θ_2 is lowered, we see that the drop in temperature of θ_1 is the same as the lowering of θ_2 for the same kilowatts lost.

If the output was assumed as proportional to the square roots of temperature differences we would then have—

$$\frac{O_2}{O_1} = \frac{\sqrt{\theta_1 - \theta_2}}{\sqrt{\theta_1 - \theta_3}}$$

or

$$\frac{O_2}{O_1} = 1 - \frac{\theta_2}{2\theta_1} + \frac{\theta_3}{2\theta_1}.$$

We see again, therefore, how futile it is trying to increase the output of a generator by means of fans,

etc., acting on the *exterior* of the insulating tubing, the effect on the output being practically negligible.

Driving ventilating ducts through the iron does not help the copper conductors at all at the higher temperatures, and fans are little use.

It appears, then, the only way to cool a generator is to apply the heat locally to the windings themselves. The only objection to this of which the author is aware is that the stresses on the windings are so enormous that they must be packed tightly into the tubes, thereby destroying all attempts to effectively cool them.

The only other method appears to be cooling by conduction from the ends of the inductors, but unless these were heavy conductors this does not appear very practicable.

It seems to the author that there is no reason why something could not be done in this way, viz. cooling the generator locally, using the ordinary methods of cooling to carry off the heat from those parts of the generator where the losses are independent of load.

As regards the efficiency of the arrangement, we have, as previously in the case of the fan :—

$$\epsilon_1 = \frac{O}{O + L + C^2R}$$

$$\epsilon_2 = \frac{kO}{kO + L + k^2C^2R + \dfrac{k^2C^2R}{f}};$$

therefore $\epsilon_1 > \epsilon_2$, if

$$L + C^2R < \frac{L}{k} + kC^2R + \frac{kC^2R}{f}$$

where $\frac{1}{f}$ is the fraction of the C^2R, stator losses

neutralized by refrigeration. The machine with the refrigerator will be more economical than a fan-ventilated machine if:—

$$\frac{kC^2R}{f} < \frac{L + k_1{}^2C^2R}{pk_1}$$

where k_1 is the output multiplier in the case of the fan, and p the fraction of the total losses corresponding to power used by the fan. There seems little doubt that the result is all in favour of refrigeration.

At the present time the design of electrical machinery is not very suitable for the application of refrigerating apparatus. Oil-cooled transformers of the static type might with advantage use them, as there is no trouble in this case with condensation. In the case of the 1500-kilowatt transformers referred to in the chapter on the "Ventilating Action of Fans," a refrigerator to neutralize 15 kilowatts would give a margin at 1½ times full load of about 30° C. rise, the original fan running all the time. The output could be very greatly increased, and the expense is very small—£200 or so.

The following figures may be of interest:—

A generator has a loss of 40 kilowatts. This is equivalent to

$$k = 0{\cdot}24 \times 40,$$

or 9·6 kilo-calories per second.

If this heat is to be carried off by air circulation, then we require

$$V = \frac{9{\cdot}6}{0{\cdot}2375 \times 1{\cdot}293 \times 20} = 1{\cdot}56$$

cubic metres per second, the temperature rise being assumed at 20° C.

Supposing a compressor of the ammonia type used with a condenser, the coils of piping could be arranged so that the air could be drawn over them. The power required would be about 15 B.H.P., and about 2000 gallons per hour of cooling water. The cost, exclusive of fan, etc., would be about £300 or £400.

It is, of course, quite a hopeless arrangement to try to neutralize all the losses—boxing in the whole generator. Here, if the losses exceed the rate of cooling, the temperature will rise, and since the ventilating action is removed altogether the value of $\frac{d\theta}{dt}$ rapidly increases.

The saving to be effected by using refrigeration is one of capital only, but in many cases this is of importance. In certain other cases it is almost a necessity, to eliminate risk of breakdown.

Meanwhile, in steam turbo-generator sets, it is little short of extraordinary to find the outputs of the electric generator for large units, like 3000 or 4000 kilowatts, limited by a temperature rise of 40° C. If it was not for this fact capital expenditure in stations with peak loads could be greatly reduced.

The author is very hopeful that refrigeration may form a useful adjunct to the generator.

Pracitcal Methods of Ventilating Armatures.

In most continuous current dynamos the armature is cylindrical, as regards the core which is hollow, and has in it ventilating ducts acting on the principle of Barker's Mill.

If this armature is revolved inside a concentric

cylinder the ventilating action would be very small if the air gap was narrow, as already mentioned on p. 29.

In such a case the ventilating action is due to the air escaping into the space between the poles of the dynamo, and can be readily examined by exploring the region by means of a fine silk thread which sets itself along the stream lines.

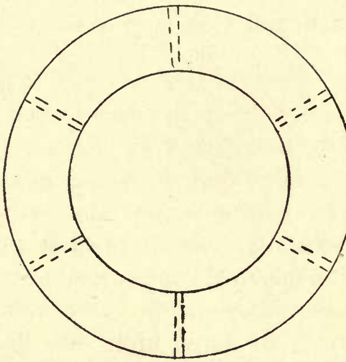

FIG. 23.

In large machines the windage may be considerable, but this appears to be more accidental than anything else. In some of the newer interpole machines the ventilation is naturally diminished.

Meanwhile it is important to get some idea of the action of the radial ventilating ducts in an armature and see in what direction improvements may be looked for.

In the case of a cylinder rotating with hollow channels representing the ventilating ducts of an armature we obviously have a forced vortex formed as in the case of Barker's Mill.

Here $$dH = \frac{v^2}{g\rho} ds + vdv.$$

If $$\rho = r,\ ds = dr,\ v = wr,\ dv = wdr,$$

$$\therefore \quad dH = \frac{2w^2r}{g}dr.$$

Hence $$\frac{dp}{G} + \frac{w^2r}{g}dr = \frac{2w^2rdr}{g}$$

or $$\frac{p}{G} - \frac{w^2r^2}{2g} = \text{Constant},$$

or $$\frac{p_2 - p_1}{G} = \frac{w^2}{2g}(r_2{}^2 - r_1{}^2),$$

p, v, r are pressure, velocity and radius; G, g are the weights of a cubic foot of air and gravitational constant respectively. The surfaces of pressure are obviously paraboloids of revolution in such a case.

In such a case if the radial velocity is required we might write

$$\frac{d^2r}{dt^2} - \omega^2 r = 0,$$

giving $$\left(\frac{dr}{dt}\right)^2 = \omega^2 x^2 + \text{Constant}.$$

If $x = a$ when $\frac{dr}{dt} = 0$ we have for the radial velocity along the channel

$$v_r = \omega\sqrt{x^2 - a^2}.$$

If, therefore, we have an armature case of thickness δ small compared with the diameter D,

$$v = 2\pi n\sqrt{D^2 - (D - \delta)^2}$$

$$v_r = 2\pi n\sqrt{2D\delta} \text{ approximately.}$$

Hence, since D is constant we see that the radial velocity of air along a duct will be

$$v \propto n\sqrt{\delta}.$$

Long ducts, then, favour ventilation for a given speed and diameter.

Now, since the diameter of an armature might be regarded as settled by centrifugal force, we might write

$$4\pi^2 n^2 D = \text{constant}$$

$$n \propto \frac{1}{\sqrt{D}}.$$

Hence the velocity of air through the armature ducts which might settle the cooling quantity of air

$$v \propto \sqrt{\delta}.$$

Since $$v = 2\pi n\sqrt{D^2 - x^2},$$

and since $n \propto \frac{1}{\sqrt{D}}$, it remains to fix the best value of x the interior diameter, in order that the cooling for a given case should be a maximum.

Since we wish to keep the interior surface as large as possible, we must have

$$2\pi n\sqrt{D^2 - x^2} \times \pi x \text{ a maximum}$$

$$2\pi n\frac{-\pi x^2}{\sqrt{D^2 - x^2}} + 2\pi^2\sqrt{D^2 - x^2} = 0$$

$$2\pi^2 x^2 - 2\pi^2(D^2 - x^2) = 0$$

$$\therefore \quad x = 0{\cdot}707D.$$

The depth of the duct then would be

$$\tfrac{1}{2}D(1 - {\cdot}7) \text{ or } \frac{3}{20}D,$$

or about $\frac{1}{6}$ of the diameter.

Another type of ventilation is where channels parallel to the axis of rotation are made use of, vanes for ventilation purposes being fixed on one side of the armature or rotor. This appears to be fairly effective, and causes a considerable draught through the rotor with consequent reduction of temperature.

Consider a vane such as shown in the figure rotating with a velocity v.

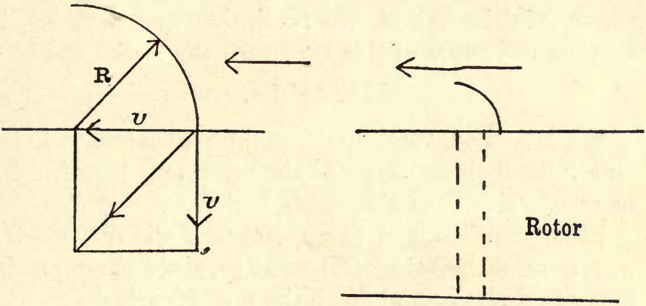

FIG. 24.

Suppose the air in front of the vane is at rest, the vane being a quadrant of cylindrical shape.

Here the intensity of normal pressure is

$$M\frac{v^2}{r},$$

and the resultant normal pressure is

$$R = 2Mv^2 \, Sin \, \frac{90}{2}$$

or

$$R = \sqrt{2}Mv^2.$$

This resultant pressure may be resolved into two components, one perpendicular to and the other parallel to the direction of motion. The latter is the force doing work on the vane. Hence

$$F = R \ Sin \frac{90}{2} = Mv^2.$$

The work done by the vane is

$$Fv = Mv^3.$$

Now v is the velocity in feet per second, say, of the rotor, so that

$$\text{work done per vane} = M(2\pi rn)^3$$

where r is the radius, n is revolutions per second. If there are N vanes of this type used the work done is

$$M.N. \ 8\pi^3 r^3 n^3.$$

We see, then, that this form of ventilation, viz. by longitudinal ducts through the rotor, may be made far more effective than a Barker's Mill type.

For a vane inclined at any angle, if v is the velocity of air and u the velocity of rotation, since these are in opposite directions, we have the relative velocity $v + u$.

The work done becomes in this case

$$Fu = M(v + u)^2(1 - Cos \ \phi)u,$$

where ϕ is the angle between the air leaving the vane, and direction of motion of the vane.

For a given vane this will be a maximum if

$$\frac{d}{du}(v^2 u + 2vu^2 + u^3) = 0,$$

$$\text{or } v = 3u.$$

That the various methods of ventilation exercise a great influence on the output coefficient is sufficiently obvious. In fact the output coefficient may be regarded as a function of the ventilation. It is an easy matter to compare the output coefficients for different ventilating systems when these are calculable.

The ordinary expression for temperature at constant watt waste W may be obtained as follows:—

Assuming Professor O. Reynolds' equation on p. 30 for cooling, it is interesting to notice that the usual expressions for output coefficients follow at once from it. We assume that the heat lost per second is carried away—

1. By ordinary convection, conduction, and radiation.
2. By rotation of the armature, the air moving over it with some velocity v depending on speed n.

$$\therefore\ W = A\pi DL\theta + B\pi D^2 Ln\theta$$

where A and B are constants, L is the length and D the diameter of the armature, θ the temperature rise of armature, n the speed.

If the ratio $\frac{L}{D}$ was known, the diameter for a given speed and temperature rise would be calculable (provided the constants were known) from the resulting equation. Otherwise we might write it

$$W = \pi DL(A\theta + BDn\theta)$$

or $$\theta = \frac{W}{k_1 + k_2 n}.$$

This is practically Mr. Esson's well-known rule.

So far as the cooling effects of air ducts are concerned, we may treat these as negligible, since the coefficient of conduction for heat across the laminations is exceedingly small. All the air ducts can do is to act as conveyors of air to the surfaces requiring cooling.

Probably one of the most effective methods of

cooling is to use tubes through which cold air or water is passed.

In such a case the theory is easily worked out as in the case of the totally-enclosed motor.

If the motor is running at a constant temperature and speed,

$$S\theta^\circ = \text{watts wasted} \times \text{constant}$$

where S is cooling surface, θ° the difference between the surface temperature and temperature of the water or cooling media.

Since $S = \pi ndl$ we have

$$\pi ndl\ \theta^\circ = \text{watts wasted} \times \text{constant},$$

and so for a given difference of temperature the number, or size of tubes, can be estimated.

Practical considerations will settle the best size to make tubes, and whether they should be corrugated or not.

Amongst the firms who are foremost in this country as regards artificial or forced ventilation the designs by Messrs. Mavor & Coulson are interesting. Mining regulations make demands for total enclosure, which means that forced ventilation must be adopted so that the machines may not be of undue cost and size.

Messrs. Mavor & Coulson's tube-cooled motors are provided with groups of tubes internal to the motor shell, and opening at both ends to the outside only. These tubes are cooled by circulating cold air through them by means of a properly-designed fan, so that the arrangement resembles a surface condenser for the heated air inside the motor case. There is no interchange of air between outside and inside the case, only the heat escaping. The motor case may be made any shape to suit requirements, and can be made absolutely dust or

explosion proof. The cables leading into the motor are led through glanded sockets in the shell.

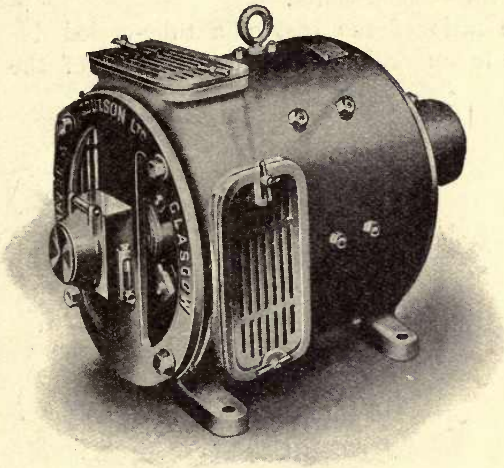

FIG 25.

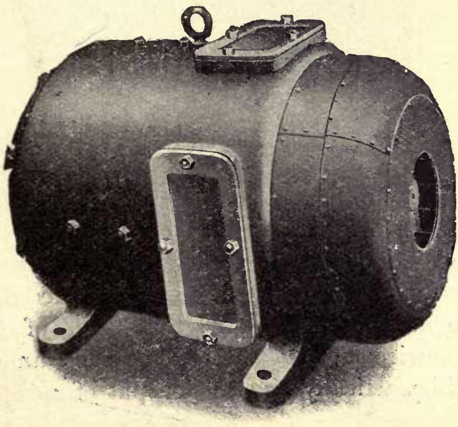

FIG. 26.

Figs. 25, 26, 27 show their ordinary ventilated motor, their tube-cooled dust-proof motor, and a three-phase tube-cooled motor.

The ratio of the cost of a tube-cooled dust-proof motor to an ordinary ventilated motor of the same

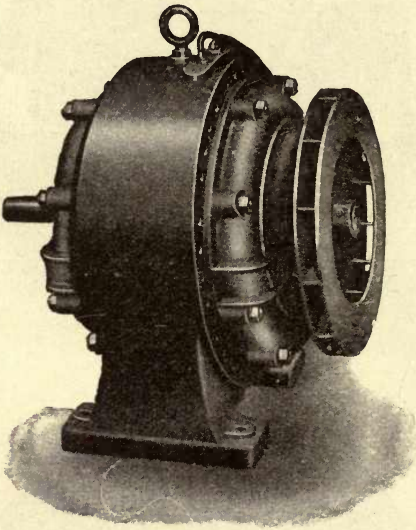

FIG. 27.

output is approximately one and a third to one and a half times.

There is no difficulty in applying the more effective water cooling in such an arrangement as this; and since we have seen on p. 23 that 90 % of the heat can be carried off by water alone such an enclosed motor will practically give the output of an open type motor.

The British Thomson-Houston Co. make various

forms of enclosed ventilated motors. Their enclosed ventilated types have a fan mounted on the armature

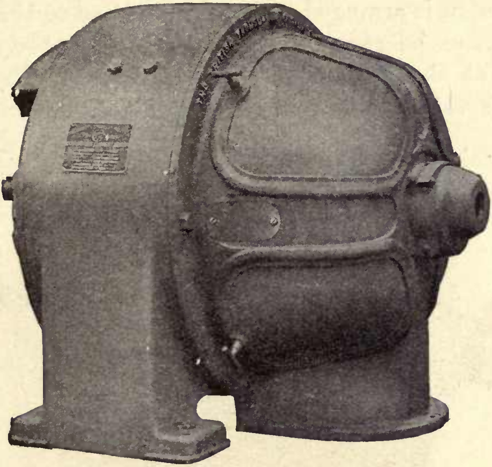

FIG. 28.

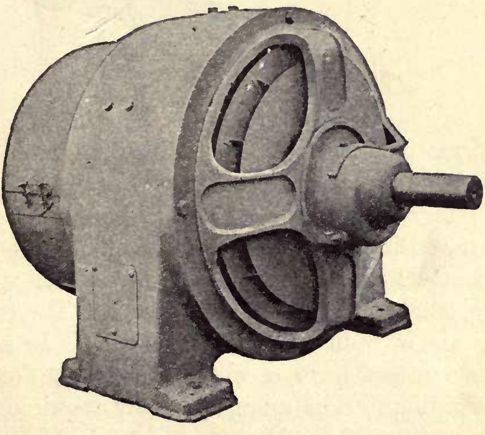

FIG. 29.

shaft which circulates the air inside the motor and discharges the warm air at the pulley end.

Their enclosed ventilated motor with fan and pipe connection is arranged with a flange fitted to the motor which can be connected to an air duct, the intake being at the bottom of the end shield. They also manufacture a rainproof construction of motor.

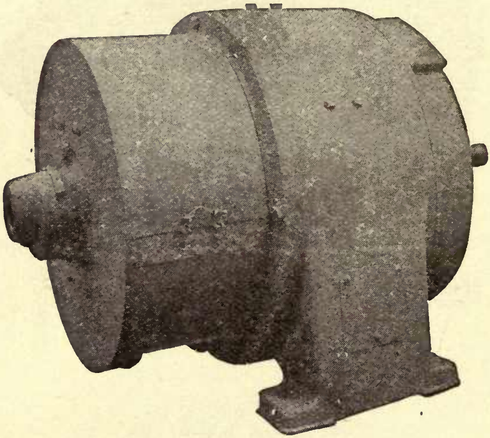

FIG. 30.

The author wishes to express his indebtedness to these firms for information and blocks of the figures shown above.

With regard to recent large turbo-generator sets, these are ventilated by means of fans on the rotor. The air passes via the rotor through the stator windings and is ejected from the top of the frame. The entrance for the air is from below the machine. In addition some generators have a water-cooled case over the stator windings. The quantity of air required to cool these generators for a given temperature rise is esti-

mated as shown in the chapter on the Refrigerator as Cooling Agent. A great deal depends on whether the air can get at the heated conductors or not.

DATA FOR THERMAL CALCULATIONS.

THE amount of heat carried off by any substance of mass M heated through θ° is

$$sM\theta$$

where s is the specific heat. The specific heats and densities of substances in common use are as follows:—

Substance.	Specific Heat.	Mass of a Cubic Centimetre in Grammes.
Air	0·2375	0·00127
Copper	0·09	8·8
Iron	0·10	7·1
"Vacuum" Oil . .	0·51	0·84

As a rule air contains a small quantity of moisture, and to find the weight of a volume V of moist air we have—

$$V \times 1{\cdot}293 \frac{1}{1 + at^\circ} \frac{H - \frac{3}{8}f}{760}$$

where a is the coefficient of expansion, t° the temperature, H the barometric pressure, f the pressure of the vapour of water in it. The presence of the water vapour adds to the thermal capacity of air, its specific heat being 0·48.

The relative humidity, and consequently vapour pressure, etc., are found by wet and dry bulb thermometer observations and a Regnault's or aluminium

cup hygrometer, and the use of Apjohn's formula, viz.:—

$$F = f - \frac{t_1^\circ - t_2^\circ}{87} \cdot \frac{h}{30},$$

for which see any treatise on Physics.

As a rule the humidity is without significance in this country, its effect on the calculation of density being only about one-half of one per cent.

Suppose a cubic metre of air contains

986 litres of dry air,
14 „ „ water vapour

at 0° C and 760 m/ms pressure of mercury. The weight would be about 11·6 grammes for the water vapour and 1276 grammes for the air. Multiplying by the respective specific heats we find a thermal capacity of 308. To heat a cubic metre of air of this composition through 1° C. requires, then, 308 calories.

This is—

$$308 \times 4{\cdot}2 \times 10^7 \text{ ergs.}$$

Since a Board of Trade unit is 1000 watts for 3600 seconds we have—

$$\frac{308 \times 4{\cdot}2 \times 10^7}{3600 \times 10^{10}},$$

or 0·03 Board of Trade units.

Suppose, then, an air stream of 10 cubic metres per second raised 1° C., then the heat per second will be—

$$308 \times 4{\cdot}2 \times 10^7 \times 10 \text{ ergs. per second.}$$

Now a kilowatt is 10^{10} ergs per second.

$$\therefore \quad \frac{308 \times 4{\cdot}2 \times 10^8}{10^{10}} = 12{\cdot}93 \text{ kilowatts.}$$

Roughly speaking, one kilowatt heats a kilogramme of dry air per second (see "The Testing of Electric Generators by Air Calorimetry," by R. Threlfall, M.A., F.R.S., *Electrician*, Nov. 27, 1893), or one cubic metre heated one degree per second requires roughly one and a third kilowatts.

$$\therefore \quad \text{kilowatts} = \tfrac{4}{3}\, Q.t^{\circ}.$$

Where Q is cubic metres per second, t° is temperature Centigrade rise.

The actual conductivity of air is very low, and can be shown to be equal to the product of the viscosity and specific heat at constant volume multiplied by a constant (see *Kinetic Theory of Gases*, Mayer). It is quite a negligible quantity so far as ventilation problems are concerned.

It is much more difficult to get heat to pass, say, from iron to air than it is to get it to pass, say, from iron to oil. Some ratios are as follows—

$$\frac{\text{Iron to air}}{\text{Iron to oil}} = 7,$$

$$\frac{\text{Iron with air blast}}{\text{Iron oil cooled}} = 5,$$

$$\frac{\text{Transformer case to air}}{\text{Oil to case}} = 3.$$

These figures give a sort of idea of the relative "heat resistance" in each case; iron to air being seven times greater than iron to oil.

In such a case conductivity is defined as—

$$C = \frac{W}{S(t_1^{\circ} - t_2^{\circ})}$$

where W are watts dissipated, $t_1^\circ - t_2^\circ$ temperature difference, S the surface. Obviously only a rough comparison is possible. Consider an armature in a totally-enclosed motor without special ventilating devices. The heat passes as follows: armature to air, air to inside of case, inside of case to outside of case, outside of case to air. There are four distinct operations, all involving air and iron.

It is somewhat to be regretted that there is apparently a connection between conductivity for heat and conductivity for electricity as in the well-known Wiedemann-Franz relation. Bad conductors of electricity are bad conductors of heat, and hence conductors generating heat are lagged with the very substance we do not want.

The following table gives the heat conductivity of a few substances. So far as the writer is aware the conductivity for heat of most of the insulating materials is unknown or unpublished.

Substance.	Conductivity.
Wood	0·00026—0·00048
Gutta Percha . . .	0·00048
Calico	0·000139
Cotton	0·00011
Paper	0·000094
Vulcanized Indiarubber .	0·000089
Vulcanite	0·000083

Of course, as already mentioned, the escape of heat from windings, etc., will depend more on the "diffusibility" or conductivity divided by the specific heat of the substance. A comparison of temperature gradients through various insulating substances might be of considerable use to designers.

A useful formula for estimating the heat loss per second is

$$K = 0{\cdot}24\, C^2 R,$$

or if there are iron losses, etc., as well,

$$K = 0{\cdot}24 \,.\, W.$$

Where W are kilowatts wasted, K is kilo-calories per second.

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